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TOBER, 1939

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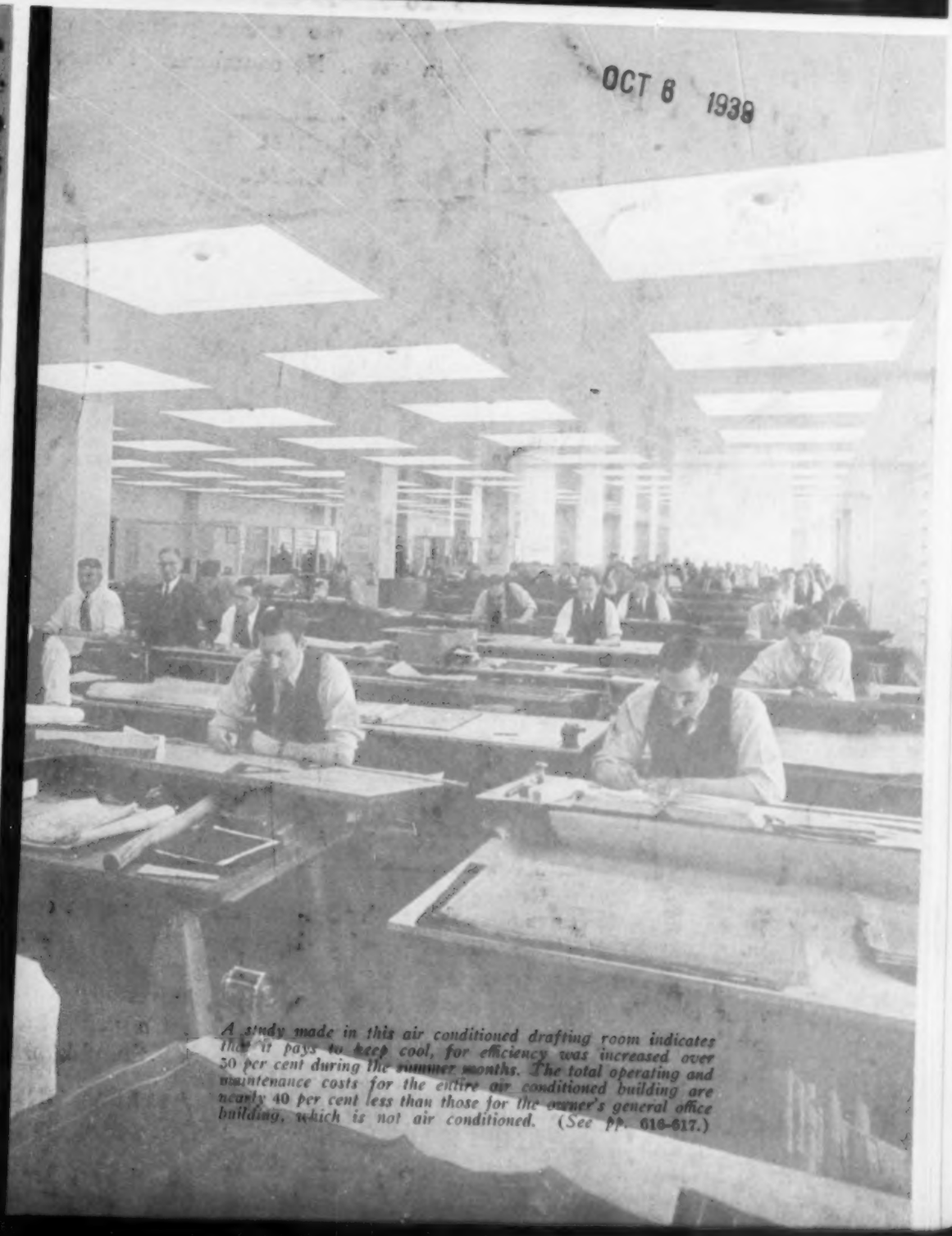


A study made in this air conditioned drafting room indicates that it pays to keep cool, for efficiency was increased over 30 per cent during the summer months. The total operating and maintenance costs for the entire air conditioned building are nearly 40 per cent less than those for the owner's general office building, which is not air conditioned. (*See pp. 616-617.*)

ABOVE: "SYLPHON" METAL DIAPHRAGM MODULATING VALVE

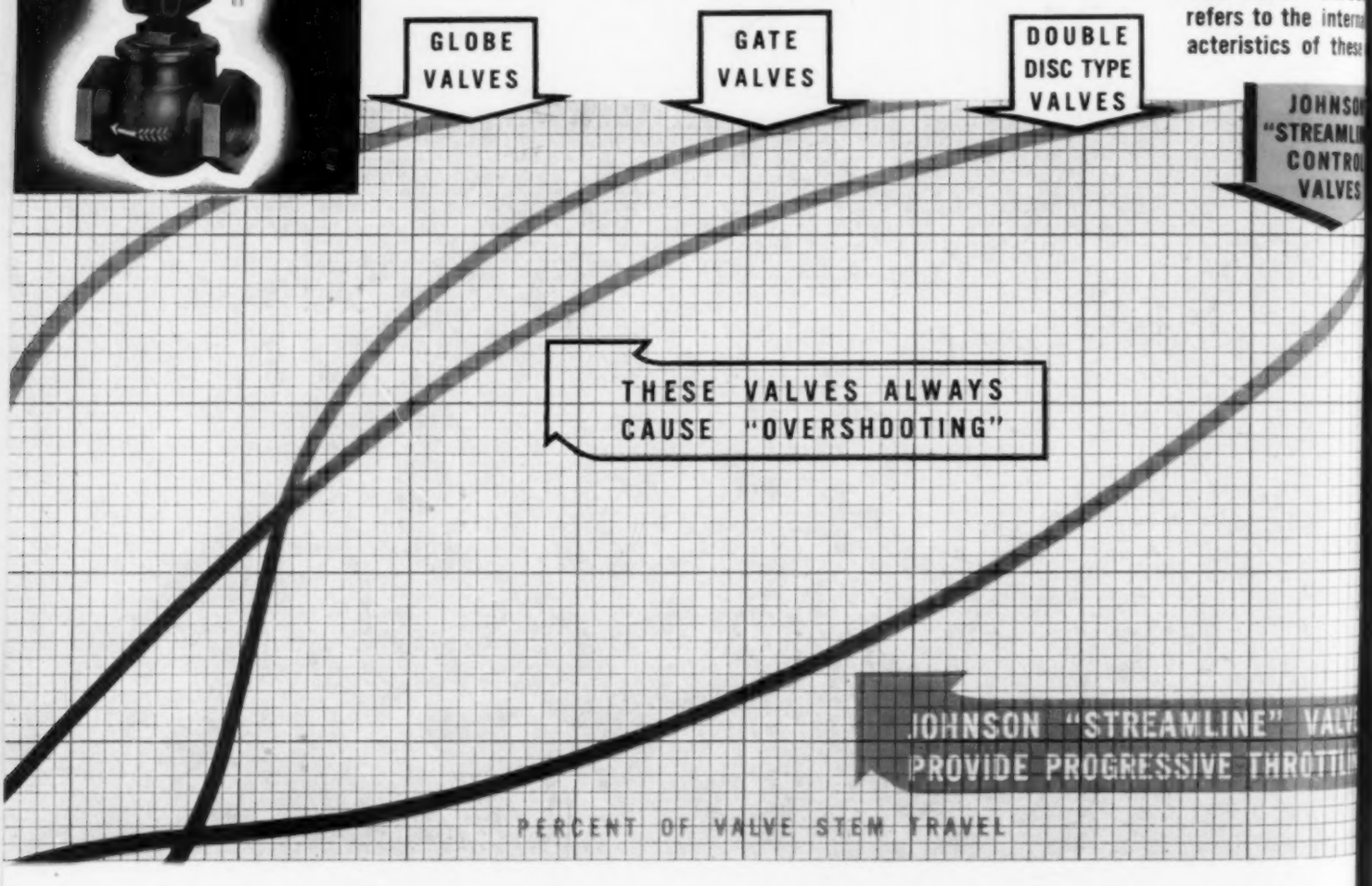
LOW: RUBBER APHRAGM VALVE WITH PILOT

GRADUAL CONTROL IS G-r-a-d-u-a-l Wh

JOHNSON "STREAMLINE"* VALVES ARE U

The partial chart below shows the impressive advancement made by Johns designing "Streamline" Control Valves. Since their introduction, several ago, the long-sought gradual action, denied in other types of valves, has possible. This engineering achievement is the result of Johnson's backg of more than fifty years in developing and perfecting automatic temperatur trol equipment . . . The Johnson "Streamline" is more than just another with a plug of special design. Even beyond the ingenious seat and disc asse the construction of the entire unit is such that perfect throttling control duced. Note the unique characteristics indicated by the curve. With Jo "Streamline" valves, the percentage of steam flow does not "jump ahea the valve stem travel. No overshooting! True gradual action!

*The word "Strea refers to the interna acteristics of these


GLOBE VALVES
GATE VALVES
DOUBLE DISC TYPE VALVES
JOHNSON "STREAMLINE" CONTROL VALVES
THESE VALVES ALWAYS CAUSE "OVERSHOOTING"
JOHNSON "STREAMLINE" VALVES PROVIDE PROGRESSIVE THROTTLI
PERCENT OF VALVE STEM TRAVEL

Johnson "Streamline" valves are available with "Sylphon" seamless metal bellows or with enclosed and protected diaphragms of super-aged and heat-resistant rubber. Direct acting (normally open) or reverse acting (normally closed) types, depending on the application . . . Rubber diaphragm "Streamline" valves are fitted, as an optional feature, with Johnson pendable pilot feature, for smooth gradual opera independent of friction and pressure variations. W pilot-operated "Streamline" valves are controlle Johnson's "adjustable sensitivity" instruments result is the ultimate in superior GRADUAL CONT

ASK FOR BULLETINS DESCRIBING "STREAMLINE" VALVES

JOHNSON SERVICE COMPANY, MILWAUKEE, WIS. • Direct Branch Offices in All Princip

The Editor's Page

■ **Many of the** manufacturers and associations of manufacturers in the heating, piping and air conditioning field have announced price policies or have made public their recommendations as affected by the war in Europe. The tenor of these statements has in general been conservative and is to be commended....J. F. Lincoln, president of the Lincoln Electric Co., for instance, pledged his company to maintain as far as possible present prices, and if materials are increased in price or the cost of labor is increased, to raise selling prices no more than the bare increase in the cost of raw materials and labor. Mr. Lincoln pointed out that other wars had brought inflation, followed by unemployment and depression after peace was achieved. ....Grant V. Wilson, president of Grant Wilson, Inc., manufacturers of insulation and fireproof materials, said in a letter to wholesalers of asbestos materials that it was his belief prices of asbestos—a supplementary war material—should advance only when actual consuming demand causes or indicates a shortage of supply, or production costs actually increase. He believes it is harmful to the industry to create through speculation false demands, artificial values, or excessive stocks and urged the buying of asbestos material as and in line with the consuming demands of wholesalers' customers. With experience during the last world war in mind, Mr. Wilson believes it is sensible to buy with relation to the ability of wholesalers' customers to consume, and to avoid if possible speculation and high prices due to a compounded artificial demand. ....Employees of the Swartwout Co. addressed a petition to the President and Congress of the United States—and to all other interested citizens—imploring that no change be made in the neutrality law unless it be to further strengthen it in restricting the goods to warring nations....The Grey advertising agency issued a mimeographed bulletin to several thousand manufacturers in the form of an open letter to salesmen advising them that when with customers to talk about their products and not about the war. Basis for the advice is that the buyer has heard about war and more war from salesman after salesman, and that if the salesman wants to talk about selling, it makes him realize that his feet are on the ground and that he has a job to do. The effect of war on business as distinguished from ordinary war talk and gossip is legitimate "selling conversation," however, the bulletin stressed.

■ **Russell G. Creviston** has been appointed to the newly created position of director of trade relations, Crane Co. announced last month. Object of the move is to enable the company to make even more effective contributions to trade activities in the industrial and heating and plumbing fields, according to P. R. Mork, Crane vice president. H. H. Simmons, advertising manager of Crane Co., was promoted to manager of advertising and sales promotion, succeeding Mr. Creviston....H. F. Horton was made general sales manager of General Fittings Co. last month. He has been with the company since it was formed in 1929. . . . . The Food Machinery Corp. has announced the purchase of the Kimball-Krogh Pump Co., a division of the Victor Equipment Co.

■ **Private industrial** concerns and equipment contractors have received nearly $1,700,000,000 in orders for purchase of construction materials and equipment and rental of equipment for use on WPA projects, Deputy Work Projects Commissioner Howard O. Hunter announced recently....Cast iron pipe and fittings represented $57,062,241 of the amount, and heating and ventilating equipment accounted for $9,789,798.

■ **Charles McDonough,** advertising manager of the Combustion Engineering Co., was elected president of the National Industrial Advertisers Association at the closing session of the NIAA's annual conference in New York City on September 22. Among the vice presidents elected is Terry Mitchell of the Frick Co., and R. L. Towne of Surface Combustion Corp. was named secretary-treasurer.

■ **Approval** by manufacturers of cast iron radiation has been given to a program which will make effective a reduction of almost 50 per cent in the number of sizes of cast iron radiators; the program has been developed and sponsored by the Institute of Boiler and Radiator Manufacturers.....Large tube radiation will be produced in 17 sizes instead of more than 30, and small tube radiation in 11 standard sizes rather than 17.

■ **The heating** and ventilating industry will receive an estimated total of 163,700 man-hours of work in the construction of a new $4,920,000 technological institute building at Northwestern University. The building is to be erected on the university's Evanston, Ill., campus with part of a $6,735,000 gift of the Walter P. Murphy foundation of Chicago which established the institute. It will be used for training a student body of 800 in civil, chemical, electrical, and mechanical engineering....Ground will be broken for the building about January 1. It will be four stories high, 500 ft long, and will contain approximately 5,500,000 cu ft. University engineers estimate the following amounts of labor will be necessary: in the shop and in the preparation of raw materials, 22,000 man-hours for heating equipment and 55,000 for ventilating; in installation of the equipment, 44,000 man-hours for heating and 42,700 for ventilating.

■ **J. H. Vogel** has been appointed general works manager of the York Ice Machinery Corp., W. S. Shipley, president, announced last month. Mr. Vogel first entered the employ of the corporation in 1915 as an apprentice in the industrial course....Lawrence E. Joseph has been made executive officer in charge of the Blaw-Knox Div. of Blaw-Knox Co., succeeding R. F. McCloskey, Sr., a company vice president who was placed in charge of development of new products for the entire corporation . . . . Frederick E. Hartmann, general sales manager of the Baker Ice Machine Co. died suddenly August 7 at his Omaha home. Mr. Hartmann joined the company in 1905—a year after it was organized—as a mechanic, and had been general sales manager for the last six years.

■ **The centennial** of the birth of Robert Henry Thurston, pioneer in engineering education and first president of the ASME, will be celebrated on October 25 at Ithaca, N. Y., by Cornell University, ASME delegates, Cornell and Stevens Institute alumni, and other groups....Dr. Thurston was an authority on materials of construction, steam engines and boilers, the history of engineering, and the philosophy of education.

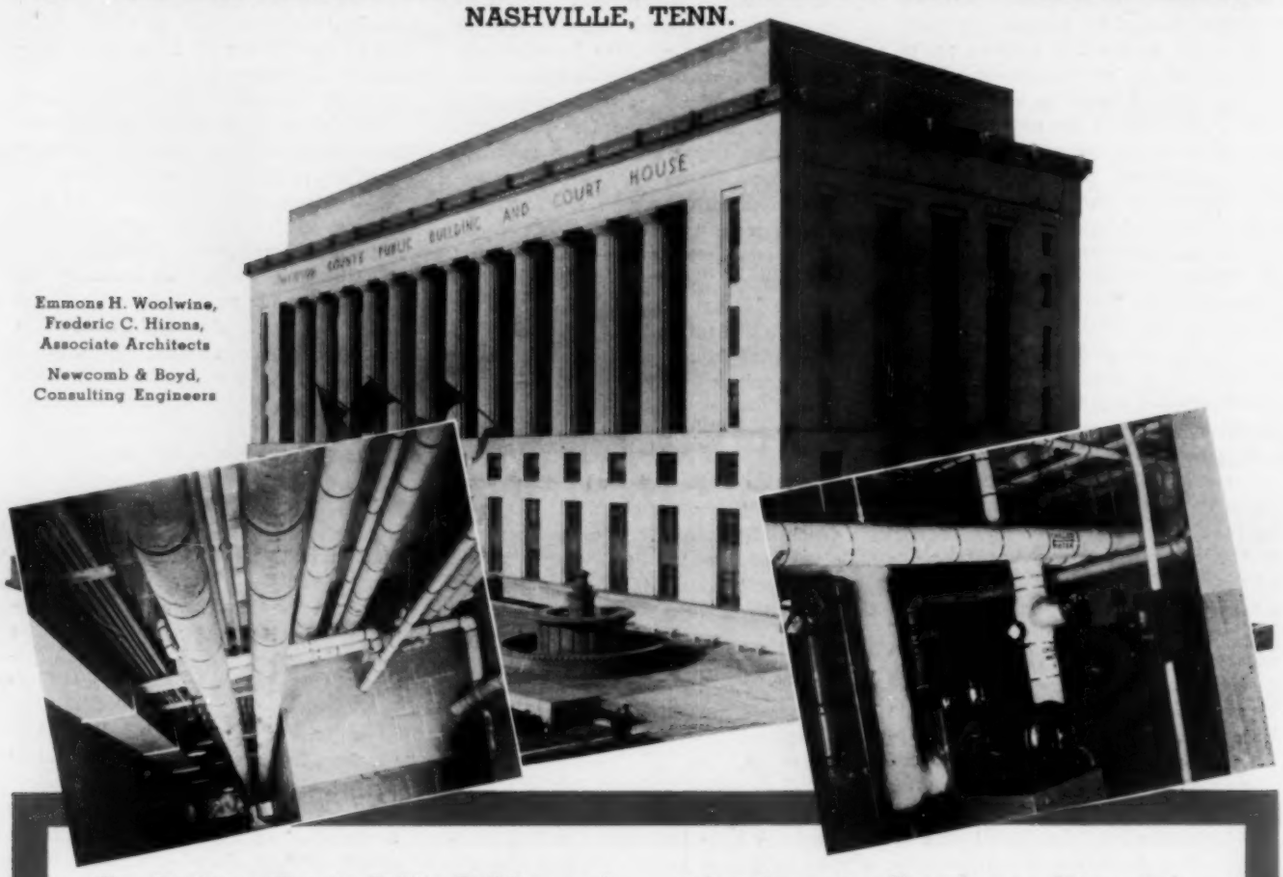
■ **It pays to think** in General Electric, for during the first six months of this year the company under its suggestion system paid $32,570 to employees who submitted new ideas. This was an increase of 4 per cent over the corresponding period of 1938.

BYERS WROUGHT IRON

USED IN 7 CORROSIVE SERVICES

IN THE DAVIDSON COUNTY PUBLIC BUILDING AND COURT HOUSE

NASHVILLE, TENN.



Emmons H. Woolwine,
Frederic C. Hirons,
Associate Architects

Newcomb & Boyd,
Consulting Engineers

The Davidson County Public Building and Court House, Nashville, Tenn., illustrates the emphasis placed on proper material selection by today's designers. Investigation of local conditions showed that corrosive attack might be anticipated in certain cold water lines; drain lines; fire lines; sprinkler lines; and gas lines. Experience data showed that wrought iron would best withstand these conditions. Byers Wrought Iron was used.

Strength of materials has always been a prime concern of the architect; durability of materials is now recognized as meriting equal consideration. Countless dollars have been saved, and will be saved, because of this new appreciation of the importance of fitting the material to the job.

The job of selecting the piping which will serve best is simplified through a service which we offer, and which many architects and engineers use. If you have a piping specification problem and will write us details, our Engineering Service Department will (1) Determine the probable corrosive conditions; (2) Relate these to similar conditions encountered elsewhere; (3) Interpret the results in the light of 75 years' experience with corrosion problems; and (4) Make recommendations . . . supported by service records. Ask, too, for a complimentary copy of our bulletin, "Wrought Iron for Piping Systems."

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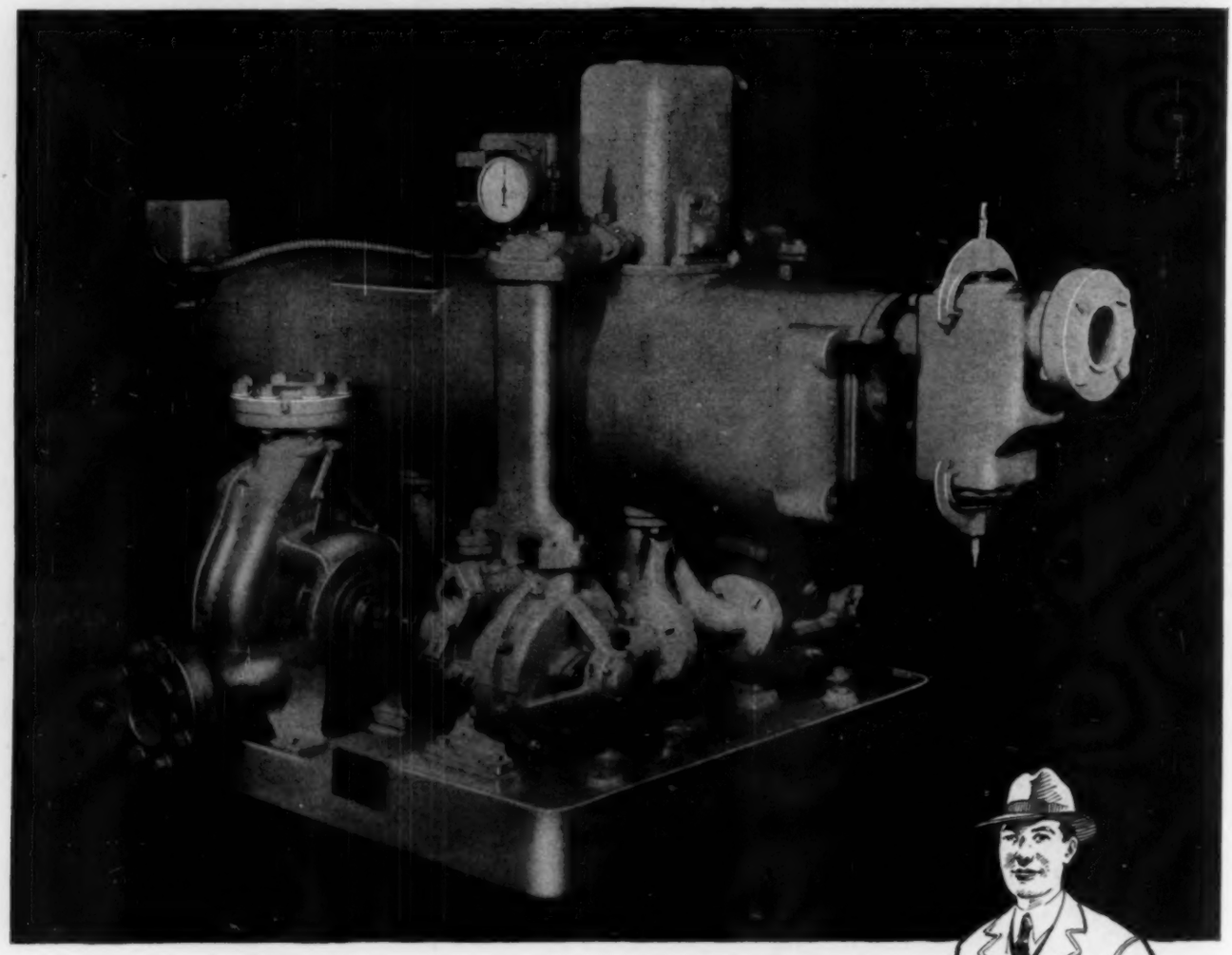
THIS ECONOMICAL HEATING PUMP OPERATES WITHOUT ELECTRIC CURRENT



The Jennings Vapor Turbine Return Line Vacuum Heating Pump requires no electric current, eliminating the one biggest expense item in the operation of an ordinary return line pump. This is because the motive power is a special turbine operating on steam direct from the heating system. Steam used to drive the turbine is returned to the system for heating, with little heat loss.

The really important saving promoted by the Jennings Vapor Turbine is in the system itself, however, for this is the only heating pump that can operate continuously with economy. Continuous operation means uniform circulation, and uniform circulation means a big saving in steam.

This pump has but one moving part, no internal wearing parts, and no internal lubrication. It is compact and quiet, and will give long and satisfactory service at the lowest maintenance cost. Bulletin No. 246 tells all about it. It is free on request.

THE NASH ENGINEERING COMPANY

SOUTH NORWALK, CONNECTICUT, U. S. A.

Heating·Piping
and Air Conditioning

Vol. 11, No. 10 • *October, 1939*

How to Plan an Operating Schedule for Heating and Air Conditioning

By Charles N. Bailey*

THE operating schedule scheme described herein can be applied to any plant that takes care of more than one air conditioned space, but becomes more useful as the number of spaces increases. However, the methods of determining the schedules can be applied to any plant—even though it conditions only one space—with possible improvements in operating economy.

The plant under consideration serves a small store and office building and consists of a 100 ton refrigerating machine and three air washers having a total capacity of 49,400 cfm of air. Each washer consists of preheat coils, filters, spray sections with a double spray bank, reheat coils and fans. In addition there are two filtering units supplying kitchens with cleaned (but not cooled) air. The temperature control system is a low voltage electric type both for the washers and the local temperature control units. The local temperature control system, which is called the "booster system," consists of unit heater-coolers in each store or office space, each operated from a local thermostat. The setting of the thermostat is left to the tenant as each has a different idea of what temperature he prefers. These booster units are on a forced water circulating system that uses hot water in the winter and chilled water in the summer. Steam for winter heating is purchased.

When the building was erected only five of the present 20 leases had been consummated and the full operating conditions of the plant could not be foreseen. For this reason a few changes have had to be made to suit the new conditions as they arose. The branches from the trunk ducts have been fitted with easily operated shut off dampers in addition to the regular volume control splitter dampers so the stores that are unoccupied need not be air conditioned. These dampers are either open or shut, depending on the tenants' needs, and are never used for volume control. This arrangement has proved to be a great aid in lowering the cost of operation of the plant because some stores open as early as 6 a. m. and others open at different hours, even as late as 10:30 a. m. Likewise the closing hours vary from 5 p. m. to 1 a. m. The air duct that takes care of the offices has not been fitted with these special shut off dampers because the office hours of the various tenants are quite similar and the savings, if any, from shutting off one or two offices a half hour earlier than the rest would be small.

The local temperature control units were originally zoned on four pipe circuits each of which has a supply and return valve. It has been found that additional control of these units is necessary, so the electric feeds to the fans have been redivided and so switched in the basement that the operator can turn off any fans in unoccupied spaces. This means that if one unit heater-cooler in a zone has to operate, the fans of the others can be turned off. Only 11 switches are needed to take care of this arrangement as several of the stores that have similar business hours are grouped.

During the first year of operation, records were kept of the inside and outside conditions and the plant was operated according to the best judgment of the engineer. A portable temperature and humidity recorder was placed in the offices and stores and simultaneous outdoor wet and dry bulb records were made with a sling psychrometer. From these records the lengths of time required for winter "warm up" and summer "cool down" were determined for the various outdoor conditions. The length of time required to prepare the space for occupancy was found to vary with temperature, day of the week (Mondays always take longer), methods used, and the degree of conditioning required within the space

Here is a description of an operating schedule and chart which has been worked out for a store and office building air conditioned the year 'round. It has been found to be valuable in aiding men untrained in the theory of air conditioning — and new men unfamiliar with the plant—to operate the system with satisfaction to the tenants and economy to the owners . . . . The scheme is applicable to any plant that takes care of more than one air conditioned space, and the methods of determining the schedules can be used to the possible advantage of improving operating economy in such plants and in plants serving only one space . . . . The procedure described covers the winter heating and humidifying cycle as well as summer operation

*Assistant to the General Superintendent, Aldis & Co., Building Management.

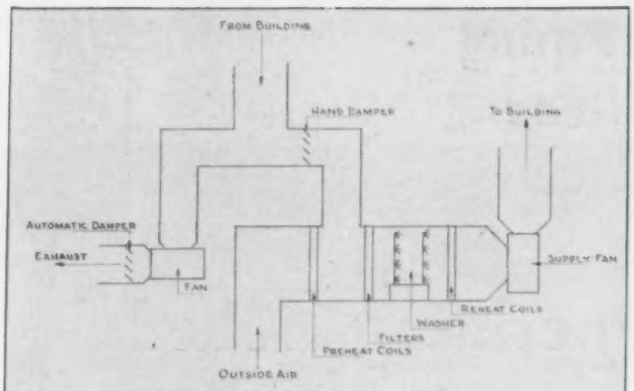


Fig. 1-Original Installation

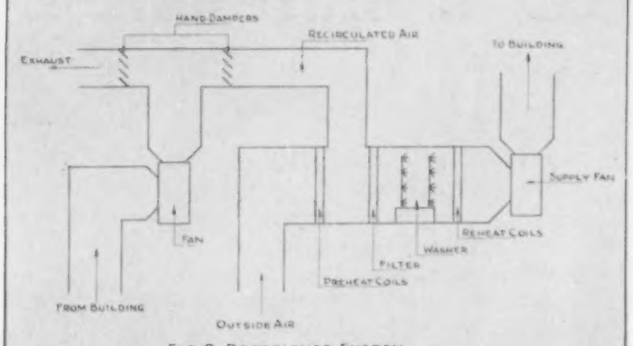


Fig. 2-Redesigned System

Figs. 1 and 2—The original duct layout was changed to prevent possibility of exhausting outside air from the preheat chamber of the air washer if there was high resistance to air flow due to the filters being dirty but not dirty enough to throw away

upon arrival of the tenant. There is but little variance in this latter, although one or two tenants want very little cooling in the summer.

The fastest method of warming up the building in the winter—and probably the most economical because of the shorter time required—is to heat the air to the highest temperature possible and circulate it through the spaces until shortly before opening time, when it is reduced to normal operating temperatures to remove the odors of overheating. During this period no outside air is taken in, but the air is kept recirculating by running both the supply and exhaust fans. Temperatures often reach 120 F and no effort is made to control the humidity until normal temperatures are again established.

In order to use both fans for recirculating the air, changes had to be made in the duct work. Originally the ducts were installed as in Fig. 1. It will be noticed that if there were high resistance to air flow through the filters, it would be possible to reverse the air flow through the recirculating duct and exhaust outside air from the preheat chamber of the air washer, thereby preventing any recirculation and reducing the economy of operation. This reversal of the air flow actually took place when the filters were dirty even though they were not dirty enough to throw away. Due to this action the useful life of the filters was shortened.

With the new arrangement of the ducts, Fig. 2, it is impossible to reverse the flow of air in the recirculation duct. However, to control the flow of air through this duct more accurately the exhaust fan discharge damper, which was automatic, has been changed to manual operation. With the exhaust fan damper and the recirculating duct damper both manually operated, any percentage of exhaust air can be recirculated at the will of the operator. This arrangement has been carried out on each of the air washers.

These dampers—with the shut off dampers, zone circuit valves and booster switches, which have all been added since taking over the plant—have not been made automatic for several reasons. First, their complex operation would require a very intricate control system, and secondly, the building caters to a large number of tenants and the normal rate of change among them must be expected; of course, with each change in tenancy, an alteration is involved which affects the services from the air conditioning plant and it is easier to adapt simple control systems to new routines than to make over a complicated unit. Thirdly, the expense of a complicated control system, with program switches and all is probably unwarranted for this type of a system. And last—but not least—the variations in the tenants' demands for heating and cooling, both as to the degree desired and the extra hours for special or overtime work, discouraged any arrangement which could not be easily changed. As a result of this, only about half of the functions within the plant that are normally automatic are so operated. The other half are manually operated.

The Operating Schedule

This large amount of manual operation requires that some sort of a schedule be arranged for the operating crew so that things will be done at the proper time and not forgotten. The schedule finally evolved is shown in Fig. 3. It will be noticed that there are four variables on this two-dimensional chart; (1) the functions to be performed, (2) the time to perform the functions, (3) changes in the time due to variations in the outdoor weather, and (4) changes in the time due to the variations in the tenants' business hours during different days of the week. The winter and summer schedules were separated because there is an entirely different group of operations for the two seasons and listing all items on one chart makes it too bulky for easy reading. The colored chips that denote the different days of the week are made of colored scotch tape, the kind used for tying Christmas packages. This tape is preferable to colored inks because it can easily be removed from the chart and placed elsewhere, or a new color substituted, and because it comes in rich colors that make a better looking chart. The ability to change the chart quickly with changes in occupancy is of great advantage.

Fig. 3 shows one of the sections of the chart used during the winter; it happens to be the one for shift No. 3, which stands watch from midnight to 8 a. m. Notice that there are four different color chips representing different combinations of the days of the week: blue is for Sunday only, red is for all week days, yellow is for Monday only, and green is for the other week days. It is impossible to use a different color for each day of the week because each of the colors would have to follow the functions that are performed every day of the week. It has made a simpler chart to use the method selected and it has not been too confusing to have two color chips representing the same day.

Let's assume the operator is reporting for work at midnight Sunday. He sees from the color code on the

chart that he is to do all functions opposite a blue chip, so he will look on the chart at the proper time, which is 12:01 a. m., and he will do all of the things listed in the first seven lines. The closing down schedule is not affected by the outdoor temperature. At 5:30 a. m. he reads the outdoor temperature thermometer and we'll say it shows a reading of 5 below zero. He again checks the color code and finds he is to do all operations followed by a red or yellow chip (Monday morning). He checks with the outdoor temperature code and finds that he is to use column No. 1 under each "time of day" heading so he looks under 5:30 a. m., follows down column No. 1 and finds the second group, of six functions, is to be done as each of them is opposite a red or yellow chip.

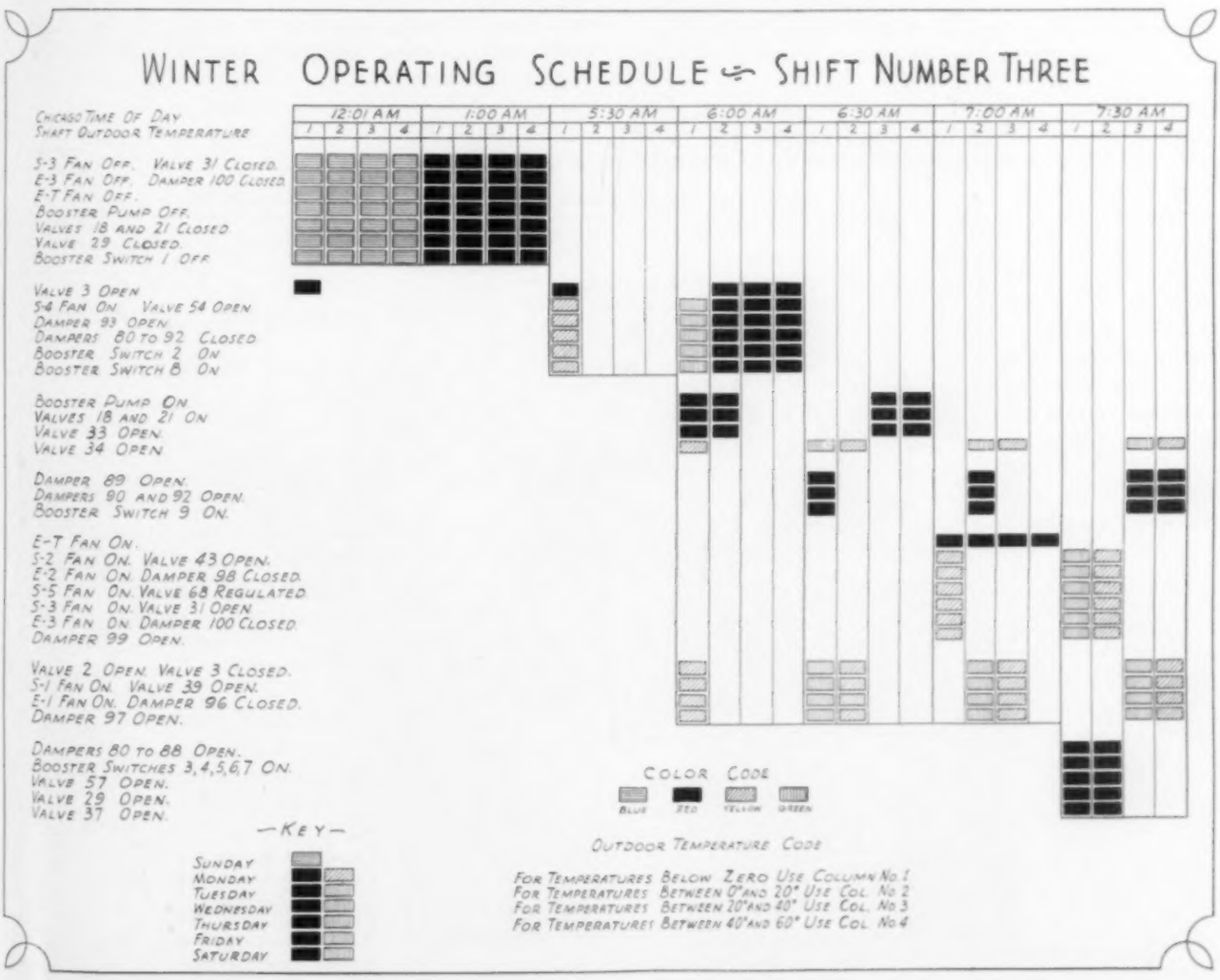
At 6:00 a. m. he again checks with the schedule and he finds in column No. 1 two more groups of four sets of functions to be done. At 6:30 a. m., when he follows down column No. 1, he finds only three things to do opposite a red or yellow chip. Those opposite the green chips he does not pay any attention to because they are to be performed only on the other week days. You will notice, however, that he has already completed these functions a half hour earlier. At 7:00 a. m. there are seven more items and at 7:30 a. m. the last five items on the chart will complete the day's work.

Notice that for each 20 deg difference in the outside temperature, there is a change of half an hour in the performing of certain duties. Also note that on Monday morning certain functions are to be done a half hour earlier than on other week days. These variations were determined from the records taken with the recording thermometer and are caused by the deficiency of heat on Sunday which allows the building to cool down more than on other days. There is very little that the operator can get mixed up on once he gets used to the color code and the temperature code.

Charts Supplement Schedule

In addition to the schedule, the operator also has an operating chart, Fig. 5. The two lower graphs on this chart tell him what settings he must make on the thermostats of the main supply fans and also the water temperature he is to carry in the booster circuits or circulating water temperature. The large graph in the upper left-hand corner is the summer temperature code and tells the operator what column, under the time of day headings, to use on the summer operating schedule. The wet bulb temperature affects summer operation so he has to use the sling psychrometer for the outdoor temperature readings. At the intersection of the outdoor wet and dry

Fig. 3—This operating schedule allows for four variables



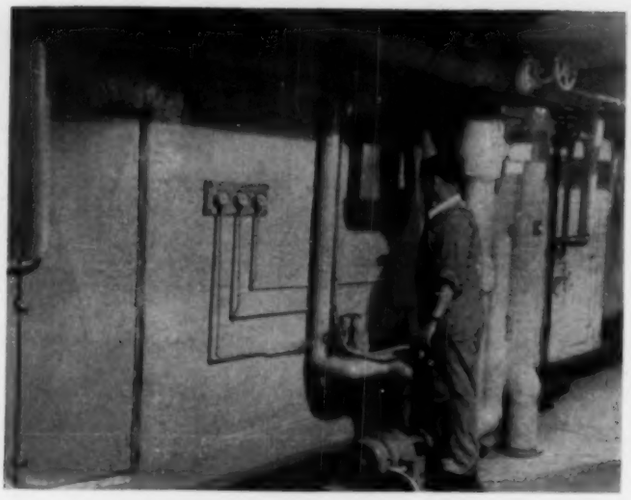


Fig. 4—Operating engineer flushing the nozzles in one of the air washers

bulb temperatures on the graph, he will find the zone marked with the column number he is to use. The zone marked *4* incidentally calls for no heating or cooling; zone 3 calls for adiabatic cooling only; zone *2* calls for refrigeration and zone 1 calls for the refrigeration to start a half hour earlier. Zone 3 is chopped off at the 58 F wet bulb line because any higher temperatures will give too high a humidity in the building with adiabatic cooling only. The condenser water rates shown are to enable the operator to check his water consumption easily through the day.

In addition to the schedule and operating chart, there is an operator's manual that sets forth the function of each unit in the engine room. The manual is illustrated with photographs as shown in Fig. 6. The items are clearly described and the methods of operation and maintenance are given. The manual describes each unit separately and tells what it is for, how it does its work and what to do if the unit fails to function properly. There is also a list of all valves, dampers and switches by number giving their location and what they control, and a list of all the valves, dampers, switches, fuses, meters, etc., arranged according to the tenants' spaces. Such a manual is invaluable to a new man entering the plant, or a man making a sick relief for a short time. It also comes in mighty handy for the regular operators to refer to.

With all this scheduling of operations, the plant has been efficiently run by men who have not been trained in the theory of air conditioning. It takes about a week for an inexperienced man to be able to do a good job of operating the plant—not that he has learned the rudiments of the theory behind it, but he has learned to interpret the schedules and charts and has gained knowledge of the location of most of the valves, dampers and switches that have to be watched. The men have followed the schedule carefully.

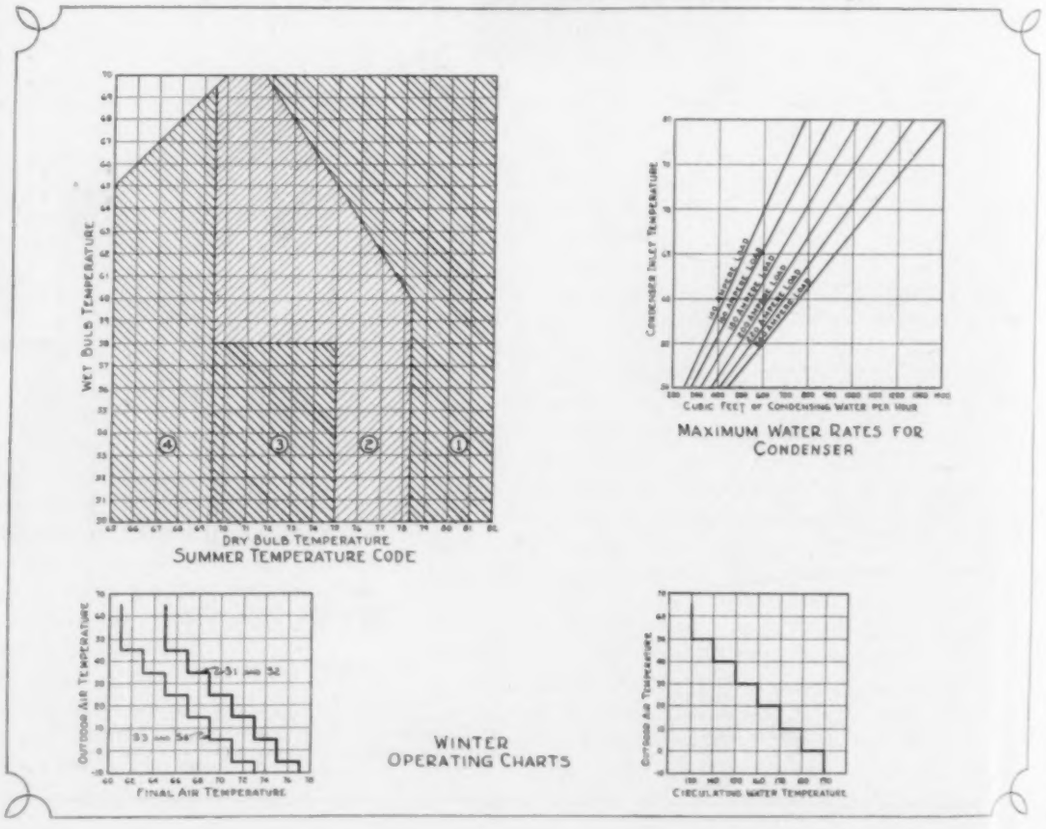
Results Show Value of Scheme

The total amount of steam used for heat last season in this Chicago building was 2,944,000 lb, and the total degree day figure was 5865, giving the economical consumption figure of 0.72 lb of steam per degree day per 1000 cu ft of building volume. The refrigeration machine has used an average of 95,640 kwhr and 960,000 cu ft of water per summer for the three full seasons it has operated. This is approximately 2.5 kwhr and 25 cu ft of water per sq ft of rentable area per summer, or a cost for both of $0.0507 per sq ft.

This cost, of course, is only for the electricity and water for the refrigeration machine. The cost of operating the fans is not included, but amounts to approximately $0.02 per sq ft more per summer. The total cost of air conditioning, including all standby charges of interest, amortization, insurance, labor etc., is of course higher, but these items are not included in the scope of the engineering department's present work.

One of the things that helps maintain economy of operation is the regulation of the recirculated air accord-

Fig. 5—This operating chart is used with the schedule shown in Fig. 3



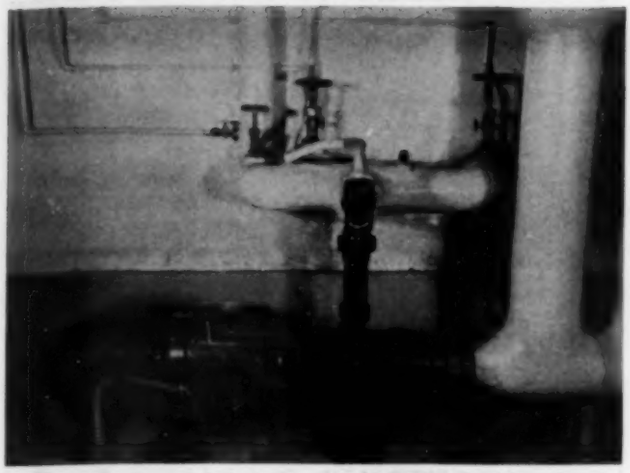


Fig. 6 shows the spray pump and its piping. On the extreme right is the steam line for humidifying and the recirculating line from the washer pan. The large valve just below the pipe shuts off the recirculated water. The next large pipe is the chilled water line from the refrigeration machine. The thermostat shown on the discharge pipe from the pump controls the automatic valve on the chilled water line to regulate the temperature of the washing water. The three smaller valves back of the thermostat are the shut off valves for the nozzles.

Fig. 6—Sample paragraph and illustration from the operator's manual

ing to the wet bulb temperatures involved. Psychrometric tables show the wet bulb temperature as the absolute criterion of the total heat in the air so the operators watch the outdoor and the recirculated air wet bulb temperatures and at any time that the outdoor wet bulb is lower than the recirculated air wet bulb, all recirculation is stopped and the indoor air is exhausted. On several borderline days during the spring and fall such a condition will arise and also occasionally it will be found more economical to exhaust the inside air when its dry bulb temperature is lower than the outdoor dry bulb because of the difference in the wet bulb temperatures. Changing the recirculation according to the wet bulb temperature instead of the dry bulb is something many engineers overlook and automatic controls do not accomplish.

Each of the main trunk ducts is connected with a draft or slant gage, and the pressure within the ducts is kept constant by changing the vortex volume control dampers on the fan inlets. This is necessary to control the air flow into the remaining spaces when part of the branch ducts are closed off with the special dampers. The same control could be accomplished with variable speed devices for the fans in place of the vortex dampers, however. Only one draft gage is used and it is connected to a manifold in the tubing from all the ducts.

A great deal of effort and time was spent in compiling the charts and manuals, but there is no doubt that the time has been well spent and it will continue to produce economy of operation as long as the building stands.

Multi-layer Construction of Thick Wall Pressure Vessels

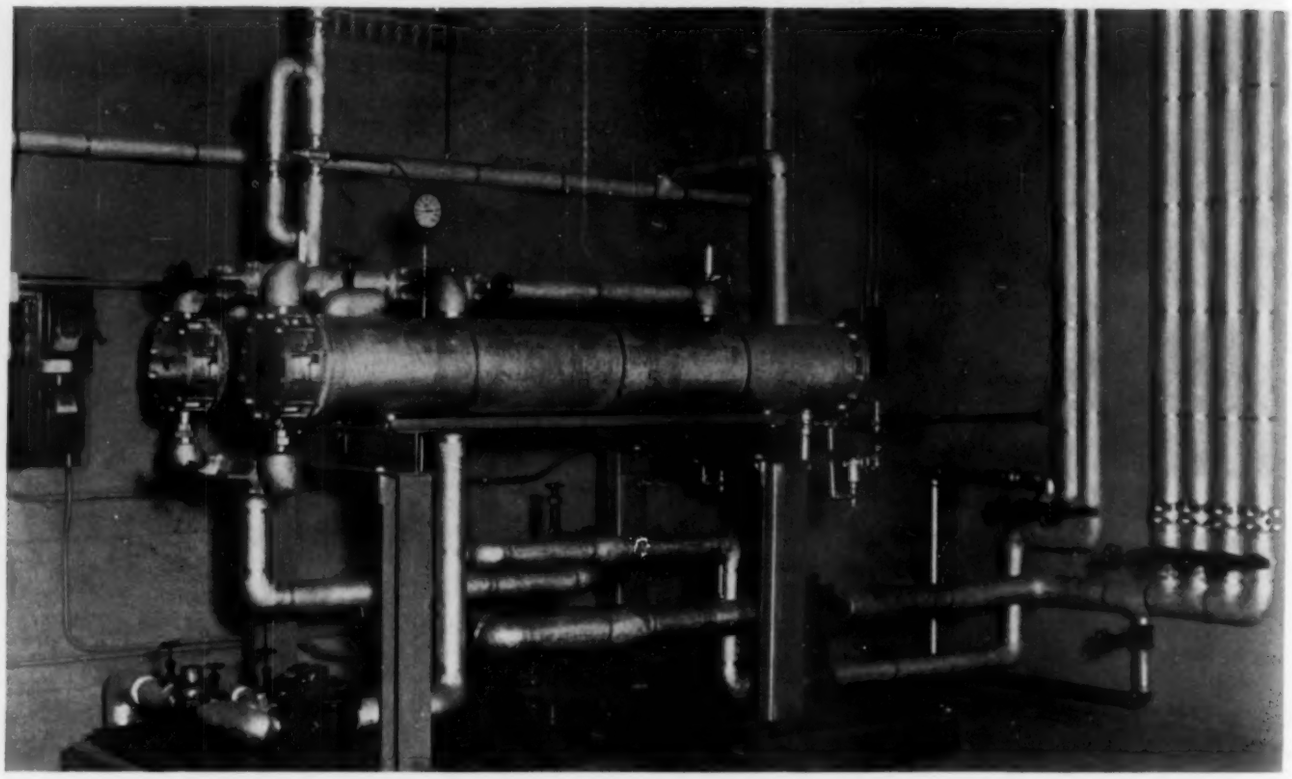
At the meeting of the American Petroleum Institute held earlier this year at New Orleans, T. McLean Jasper* discussed the multi-layer construction of thick wall pressure vessels, from which the following abstract has been prepared.

Multi-layer construction is a development which has grown out of the failure of small thick wall, high pressure vessels to give the service expected of them. Four such thick wall pressure vessels, 12 ft long, with an outside diameter of 33¼ in., and a total wall thickness of 3⅝ in., were built several years ago and tested to destruction. In no case was the pressure value at failure more than 75 per cent of what had been expected according to commonly-accepted cylinder formulas. A careful study of the test data thus gathered made it apparent that the accepted formulas were not applicable for heavy, solid wall "small diameter" vessels, i. e., vessels in which the ratio of diameter to wall thickness (D/t) was in the neighborhood of 10 or less.

This situation acted to spur on a thorough investigation, and multi-layer construction showed a marked superiority over other proposed methods. A patent was secured covering this construction. The multi-layer vessel is constructed of several layers of descaled steel, each progressively wrapped and tightened around the cylinder by mechanical means and then welded together at the edges. The thickness of each outside layer is generally less than ¼ in., whereas the inside layer may be heavier. Only the inside layer is leakproof. Each subsequent layer is perforated with small holes for venting. After all layers have been added, the ends are machined for the welding groove. The contact between layers is so uniform that it is difficult to determine from visual inspection that the vessel is not made from solid steel plates. With the ends machined, the sections are assembled by welding in a manner similar to the way sections of solid wall vessels are joined.

The tight fit between various layers is apparent from this close-up of a multi-layer shell

*A. O. Smith Corp.



The oil pumping, heating and straining equipment may be neatly and efficiently arranged on a steel pan and delivered to the job as a unit

Piping and Handling Heavy Fuel Oils for Boilers and Industrial Furnaces

By Paul R. Unger*

HEAVY fuel oils have been widely used for large boiler plants in both stationary and marine practice and also for large industrial applications of various types. In the past few years, however, a new series of problems has arisen that had occurred previously in rather isolated cases. The medium sized fuel user in both commercial and industrial plants has evinced a great interest in burning heavier fuels as the result of a differential in prices between light and heavy fuel oils throughout the country, the natural desire for low cost fuel operation, and the rapid improvement in equipment. The demand has been not only to burn this residual oil but to burn it with practically automatic controls. We can probably best discuss the subject by dividing it into classifications and discussing each one separately, afterwards discussing such plants as may be a combination of these classifications.

Our first classification will consider low pressure steam and modern forced hot water boiler plants; the second will deal with high pressure boiler plants; the third will cover industrial furnaces and processes of various kinds; and the fourth will treat of plants having a combination of boilers and industrial furnaces under the same roof.

The principal problem involved in burning Bunker C oil is one of transportation—bringing the oil from the storage tank to the burner at a temperature which will reduce the viscosity of the oil to where it will flow freely and atomize easily. At first glance, this may seem easy. One might think that all that is necessary is to put a steam coil in the storage tank to keep the tank warm. This, of course, is a step in the right direction and is exactly what was done in the early oil burner installations in large boiler plants. Heavy fuel oils, however, require a higher temperature for actual atomization than for pumping the oil from the tanks. Assuming, therefore, that a temperature of 170 F is necessary for combustion it is obviously wasteful to keep the entire tank heated to 170 F. Not only is it wasteful, but for the average oil burning installation some 50 to 60 ft away from the storage tanks, there is also a considerable temperature drop before the oil gets to the burners, which means that a still higher tank temperature would have to be maintained. This also introduces serious problems in pumping the oil to the burners as due to the vacuum

*Mid-West Heat Service Co., Combustion Engineers.

in the oil suction line created by the fuel oil pump, the flash point temperature of the oil would be lower and cause gassing, which seriously affects the maintenance of a steady oil pressure. Therefore it is both economical and advisable to maintain the oil in the storage tank at as low a temperature as is conducive to good pumping conditions.

Generally speaking, the oil pumping temperature for Bunker C fuel oil is approximately 100 F. The steam coils originally used for this purpose consisted of continuous pipe running back and forth the full length of the tank until sufficient heating surface was introduced. This, of course, maintained a high temperature in the entire oil storage tank. Modern practice has shown that this is not necessary. It is necessary to maintain the high temperature of the oil only locally at the point where the oil enters the oil suction stub. This is accomplished by various methods:

1) Using a spiral heating coil with the suction stub connected in the center of the coil. This may be still further localized by dropping a sheet metal housing around both the oil suction stub and the coil, leaving this housing open from the bottom of the tank to approximately 6 in. above the bottom of the tank (Fig. 1).

2) The use of a suction bell which consists of a small container at the bottom of the tank into which both the oil suction and return lines enter and which contains a small coil (Fig. 2).

3) Omitting the coils in the oil storage tank entirely and recirculating the oil which is heated externally back to the oil storage tank in the oil return line, terminating this line adjacent to or in the center of the suction stub (Fig. 3). This method will be discussed more fully.

In many low pressure plants the use of the steam coil is impractical because of the difficulty in returning the condensate to the system. With the rapid development in forced hot water heating, it is natural that we should solve this problem by the use of hot water coils in place of steam coils in many low pressure installations and certainly in all installations with forced hot water boilers. Such a hot water coil consists of a spiral coil similar to the steam coil except that additional heating surface is necessary as well as a circulating pump sending the boiler water through the coils in the oil storage tank and returning it to the heating system. This circulating pump may easily be made automatic from an aquastat in the oil suction line and oil suction temperatures may be maintained within close limits.

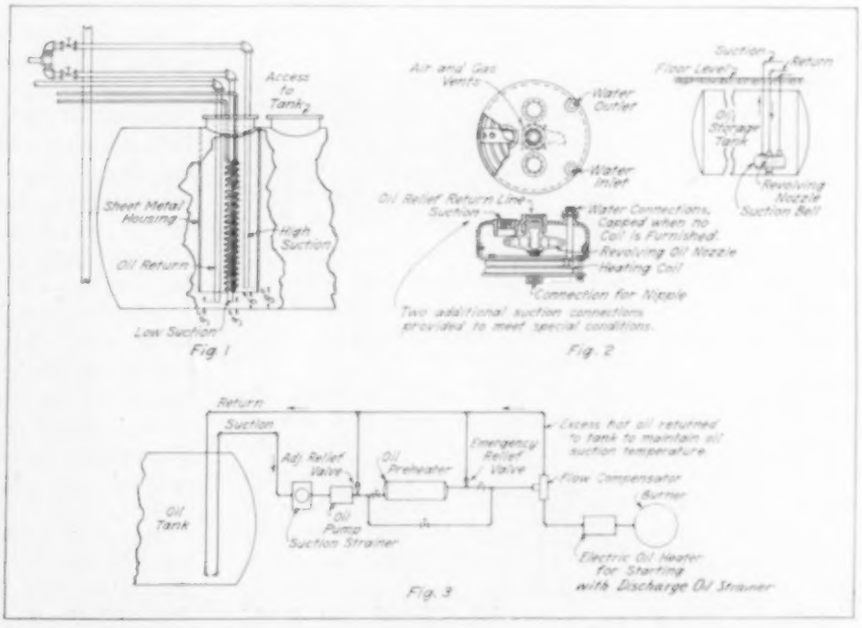
In some plants there is objection to circulating the boiler water directly into the coils of the oil storage tank because of the possibility of getting oil into the boiler in the event of a leaky coil. In these cases, a transfer heater may be used to heat the hot water for circulation with either boiler water or low pressure steam, as the case may be.

In single tank installations burning heavy oil it is better to use two oil suction lines. One of these lines, called the "low suction," originates 3 in. from the bottom of the tank. The second line, the "high suction," originates about 8 in. from the bottom of the tank. These two lines should be cross-connected in the boiler room. With such an arrangement (Fig. 1) operation is normally accomplished with the low suction stub. When sediment accumulates and operation becomes difficult it is then a simple matter to switch to the high line and use this until such time as it is convenient to clean the tank or pump out the sediment. If one suction line only were used and this happened when steam was important a shutdown might be necessary.

Mr. Unger discusses here the methods of pumping, heating, and piping heavy fuel oil to burners in medium size boiler plants—both low and high pressure. In a second article, he will consider handling heavy fuel oil for industrial furnaces and processes, and combination installations where both boilers and industrial process furnaces are fired with the same grade of fuel

Where duplicate pumps are provided it is possible to arrange the entire system so that sediment may be pumped from the tank to the sewers with one pump while the other pump is in operation from the high suction line supplying the oil burners. If the expense of two oil suction stubs and one return line is excessive it is possible to use one suction stub and one return line, running one of the lines to the low suction position and the other to the high suction position and arranging the oil pumps so that the two lines may be reversed and the return line used as a suction stub. This has been done in many cases but entails a more complicated system of valving and is not as generally satisfactory as the first method.

Where more than one boiler is used in a single plant it is usually well to provide oil pumping, heating, and straining equipment in duplicate, either set of which is



capable of supplying all of the fuel oil required for all of the burners. This consists essentially of a duplicate oil suction strainer designed so that either basket may be cleaned while the other is in service, two motor driven fuel oil pumps operating at no more than 300 rpm, two fuel oil heaters using either steam or hot water and a duplicate oil discharge strainer. Individual relief valves should be placed on each fuel oil pump and each fuel oil heater.

Where the plant is of the type that may shut down at night or over week-ends, an electric heater should be provided. The electric heater may be one unit, large enough to take care of all of the oil burners, but if the burners are to be entirely automatic or intermittent in operation, there must be an electric heater on each burner. It is possible to arrange the oil pumping, heating and straining equipment very neatly and efficiently on a steel pan which may then be shipped to its destination complete with all interconnecting piping, valves and fittings, as a unit. In this way the entire oil handling system may be tested before delivery to the job.

The oil preheaters and pumping unit should be of sufficient capacity to raise the oil to the most efficient operating temperature. This will vary with different types of oil burners, and different viscosities of oil, from 160 to 200 F. After heating the oil to the proper temperature, it is essential to get the hot oil to the oil burner as quickly as possible to minimize the drop in oil temperature between the oil heaters and the oil burners. Where burners are ignited manually it is best to take a circulating line from a point adjacent to each burner and run it back into the oil return system so that this line may be opened to purge it of cold oil that may lie between the oil heater and burners before lighting off the fires. This will eliminate a great deal of difficulty which is encountered due to the so-called "dead end" oil lines. If possible, it is well to use a continuous circulating system, circulating hot oil by the burners and back into the return line. In this event, both the heaters and fuel oil pumps would have to be larger than the actual fuel oil requirements. The writer has made a number of installations in which this type of system is used and in which the heaters and fuel oil pumps have been enlarged to a point where sufficient fuel oil is returned from the burner to the storage tanks to maintain the proper tank temperature. In many cases this has eliminated the use of coils in the oil storage tank, as discussed above. This method of heating an oil tank may be used only where the plant consists of a single oil tank. In the case of two or more oil tanks it is obvious that while returning hot oil would keep the tank in use at the proper temperature, the other tanks, lying idle, would cool down to where pumping would be difficult or even impossible. Therefore, where there are multiple tanks, coils must be used.

Where the burners are operated under automatic intermittent control, the coils in the tank and the oil heaters on the line still will not provide hot oil at the burners after an off period and improper combustion results until the heaters reach the correct temperature and the cold oil that lies between the heaters and the burners is consumed. On this type of installation, therefore, it is necessary to install an electric oil heater as close to the burner nozzle as possible. The heater should be equipped with a double circuit thermostat or two thermostats, one arranged to keep the burner from starting until the proper oil temperature has been reached and the other set to cut off the electric heater as soon as hot oil is received from the steam or hot water oil heater.

The problems in firing high pressure boiler plants with heavy oil are, with a few differences, similar to those heretofore discussed. In this type of plant it is not necessary to resort to hot water coils or hot water type fuel oil heaters. The coils in the oil storage tank should be operated at a pressure of from 50 to 100 lb. If the boilers operate at higher pressures than these it is advisable to put a pressure reducing valve in the steam line to the coils. Temperature regulation of the oil in the storage tank may be accomplished by various types of temperature regulators situated either in the suction line or in the oil storage tank actuating a steam valve in the line to the tank coils. Where high pressure steam is used for the final heating of the oil it is essential that these booster heaters also be equipped with thermostatic control as it is possible to overheat the oil.

In a high pressure boiler plant the problem of circulating hot oil from the oil pumping system to the burners is a more serious one than in a low pressure plant because the distance between the pumping equipment and the burners is sometimes much greater than in a low pressure plant. The method of piping up the oil burners from the pumping system to the boilers depends considerably on the type of oil burners used. Where rotary burners or air or steam atomizing burners are used, circulating lines are provided as close to the burner as possible and all piping may be standard weight. Where the mechanical pressure atomizing oil burner is used circulating lines should be provided at each boiler. There is at least one oil burning system on the market today which maintains continuous circulation of a uniform amount of oil at a constant pressure through the gun of the burner and through the atomizer, returning to the storage tanks after atomization that oil which is not burned. With this type of lay-out, the element of temperature drop between the heaters and the burners is practically eliminated. The inlet pressure of this system is 300 lb and the control is accomplished through a control valve on the return line by varying the back pressure between the atomizer and the control valve. This permits constant circulation, washes the atomizer and tips during operation and keeps the tips cool, thus preventing the formation of carbon. Better still, it permits operation over a very wide range of load conditions without changing tips or changing the character of the fire.

It is good practice in both high and low pressure boiler plants to insulate the hot oil lines. It is an economy measure and it helps maintain oil temperatures. Where coils in the oil storage tank are used, it is wise to wrap oil and steam or hot water lines together, between the oil pumping set and the oil storage tank. If these lines are underground they should be run in properly drained tile.

[*A second article will discuss handling heavy fuel oil for industrial furnaces and processes, and for combination installations where both boilers and industrial process furnaces are fired with the same grade of fuel.*]

Study of Building Heating Economies Shows Money Value of Good Operation

By Earle Shultz* and J. C. Butler†

FOR the purpose of determining the extent to which one can make economies in the use of steam for heating buildings and of determining which of the many methods of accomplishing these economies gives the best results, a joint study was undertaken by the National Association of Building Owners and Managers and the National District Heating Association. To secure the necessary information, a questionnaire was prepared and distributed by the two associations. Unfortunately, the work was started late in the year, and the number of replies was not as great as desired. Consequently, this must be in the nature of a progress report.

However, sufficient replies were received to furnish some general conclusions of interest and to indicate that a more complete study should be extremely valuable. One hundred and eighty-nine questionnaires were returned, of which 153 were sufficiently complete to be usable. Those that could be used were divided into two groups; 79 contained information as to steam consumption, both before and after the application of methods of heat control, while 74 gave only current data. We have not attempted in this report to analyze coal figures, because the number of returns containing this information is insufficient to yield definite results.

This report covers a study of the steam used for space heating only. Steam used for cooking, hot water and other non-space heating uses has been eliminated. As a number of replies did not separate heating steam from total steam, it was necessary for the committee to estimate this distribution. Where the consumption for the three summer months was given, this division has probably been made with reasonable accuracy. The necessity of estimating these figures, however, introduces some element of uncertainty in the results.

The questionnaire was in four parts: Part 1, statistical data; Part 2, methods of saving heat without use of control apparatus; Part 3, control apparatus; and Part 4, general conclusions. The committee's work in preparing this report has been principally that of summarizing the information furnished in the questionnaires. All opinions expressed or conclusions reached are based upon this information.

Table 1—Heating Steam Consumption Before and After Application of Heating Economies

No. of Bldgs.	Volume 1000 Cu Ft	Steam for Heating 1000 Lb		Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	Reports
		Before	After	Before	After	Before	After		
79	261,888	1,160,254	818,939	202,268	151,921	0.772	0.580	24.8	Complete
74	208,246		648,441		132,546		0.637		Partial
153	470,134		1,467,380		284,467		0.605		Total

Table 1 gives totals of the data from the two groups of questionnaires that were usable. The 153 buildings reporting contain 470,134,000 cu ft of volume, representing approximately 24,000,000 sq ft of rentable area. Their steam consumption for heating purposes at the present time amounts to 1,467,380,000 lb. More than half of the buildings returned complete questionnaires, giving figures before and after the application of heating economies. The remaining questionnaires were partially complete, giving only current figures.

The buildings giving complete figures (see Table 1) have an average steam consumption, before the application of heating economies, of 0.772 lb of steam per degree day per 1000 cu ft of volume, and a consumption of 0.580 lb of steam after the application of heating economies. This represents a saving of 24.8 per cent. The buildings giving only partial replies have a current steam consumption of 0.637 lb of steam per degree day per 1000 cu ft of volume. This is close to the figure for economical operation as calculated from the complete returns. The conclusion may be made that these buildings are largely practicing heating economies but do not have figures available which would show the amount of the savings accomplished.

Economical Rate of Steam Consumption

In preparing the questionnaire, the committee had in mind the determination of the percentage of savings pos-

To determine the extent to which economies in the use of steam for heating buildings can be made, a joint study was recently made by the National Association of Building Owners and Managers and the National District Heating Association. In brief, the following important conclusions may be drawn from it: (1) Careful and intelligent supervision of the use of steam is the first requisite in securing economies. (2) Steam meters should be installed on the supply to the heating system and daily pounds of steam per degree day records kept and compared with the standard. (3) The heating system should be maintained in good repair and if necessary overhauled so as to permit rapid circulation of steam at low pressures. This may involve the use of accurate and sensitive reducing valves and the installation of orifices in risers and radiators. All building air leaks should be eliminated. (4) A heating schedule, carefully worked out for each individual building, should be used. (5) Automatic heat control devices will definitely assure savings that will pay their cost in from two to four years

*Vice-president, Illinois Maintenance Co., and Chairman, Joint Committee on Heating Economies of the National Association of Building Owners and Managers and the National District Heating Association.
†Chief Operating and Commercial Engineer, Illinois Maintenance Co.

Table 2—Reports Grouped by Cities

City	No. of Bldgs.	Volume 1000 Cu Ft	Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	City Eff. Per Cent	Reports
			Before	After	Before	After			
Chicago	23	120,017	100,165	70,357	0.835	0.586	29.8		Complete
	13	76,553		47,703		0.623			Partial
	36	196,570		118,060		0.601		99	Total
Pittsburgh	7	28,816	18,771	14,115	0.652	0.490	24.8		Complete
	8	31,476		20,308		0.645			Partial
	15	60,292		34,423		0.571		95	Total
Detroit	13	31,525	20,569	18,464	0.652	0.586	10.1		Complete
	3	13,299		5,280		0.397			Partial
	16	44,824		23,744		0.530		88	Total
Portland, Ore.	3	6,420	5,810	4,950	0.905	0.771	14.8		Complete
	18	20,914		14,975		0.716			Partial
	21	27,334		19,925		0.729		121	Total
New York	3	26,349	15,140	11,510	0.575	0.437	24.0	72	Complete
Baltimore	6	14,135	9,264	6,607	0.655	0.467	28.7		Complete
	3	8,777		5,800		0.661			Partial
	9	22,912		12,407		0.542		90	Total
Seattle	6	19,862		8,210		0.413		68	Partial
Indianapolis	2	2,835	4,280	3,180	1.510	1.122	25.7		Complete
	7	10,904		8,945		0.821			Partial
	9	13,739		12,125		0.882		146	Total
St. Louis	5	11,891	10,001	8,565	0.841	0.720	14.4		Complete
	1	1,335		1,090		0.816			Partial
	6	13,226		9,655		0.730		121	Total
Milwaukee	6	12,733	11,633	9,218	0.914	0.724	20.8	120	Complete
Philadelphia	1	2,060	2,190	1,520	1.063	0.738	30.6		Complete
	1	7,973		3,500		0.439			Partial
	2	10,033		5,020		0.500		83	Total
Peoria	5	6,280		7,340		1.169		193	Partial
Omaha	2	2,137	1,990	1,500	0.931	0.702	24.6		Complete
	3	2,895		2,225		0.769			Partial
	5	5,032		3,725		0.740		122	Total
Cincinnati	2	3,547		3,080		0.868		143	Partial
Atlanta	1	2,717		2,720		1.001		165	Partial
Rochester, N. Y.	3	1,765	1,575	1,235	0.892	0.700	21.5	116	Complete
South Bend	3	1,714		1,370		0.799		132	Partial
Yakima	1	720	430	420	0.597	0.583	2.4	96	Complete
St. Joseph, Mo.	4	485	450	280	0.928	0.577	37.8	95	Complete

sible through modern methods of heat saving. A study of the replies, however, indicates that the use of a "percentage of saving" figure is probably misleading. The largest percentage of saving accomplished occurs in buildings starting out with the poorest utilization of steam, and in many of these (although a large saving has been accomplished) the rate of steam consumption is still high. Consequently, the committee thinks that a safer guide to efficient operation will come from the determination of an economical rate of steam consumption rather than a percentage of saving. A study of the questionnaires indicates that such an economical rate is 0.6 lb. of steam per degree day per 1000 cu ft of building volume. This figure will, of course, be subject to adjustment for individual buildings based upon their type of construction, size, exposure and occupancy. It may also be further adjusted if the committee, during the coming year, obtains steam consumption figures for a much larger number of buildings. Nor should it be considered the minimum possible, as it is the average of all buildings reporting—both good and bad. However, we believe that if every building will check its steam consumption against this figure of 0.6 lb of steam, it will be a fairly safe guide to efficient operation.

Analysis by Cities

In analyzing the questionnaires, the committee grouped the figures under several classifications—such as cities, source of steam supply, type of occupancy, type of heating system, type of control, volume of building, and hours of heating—in order to make comparisons in the several classifications.

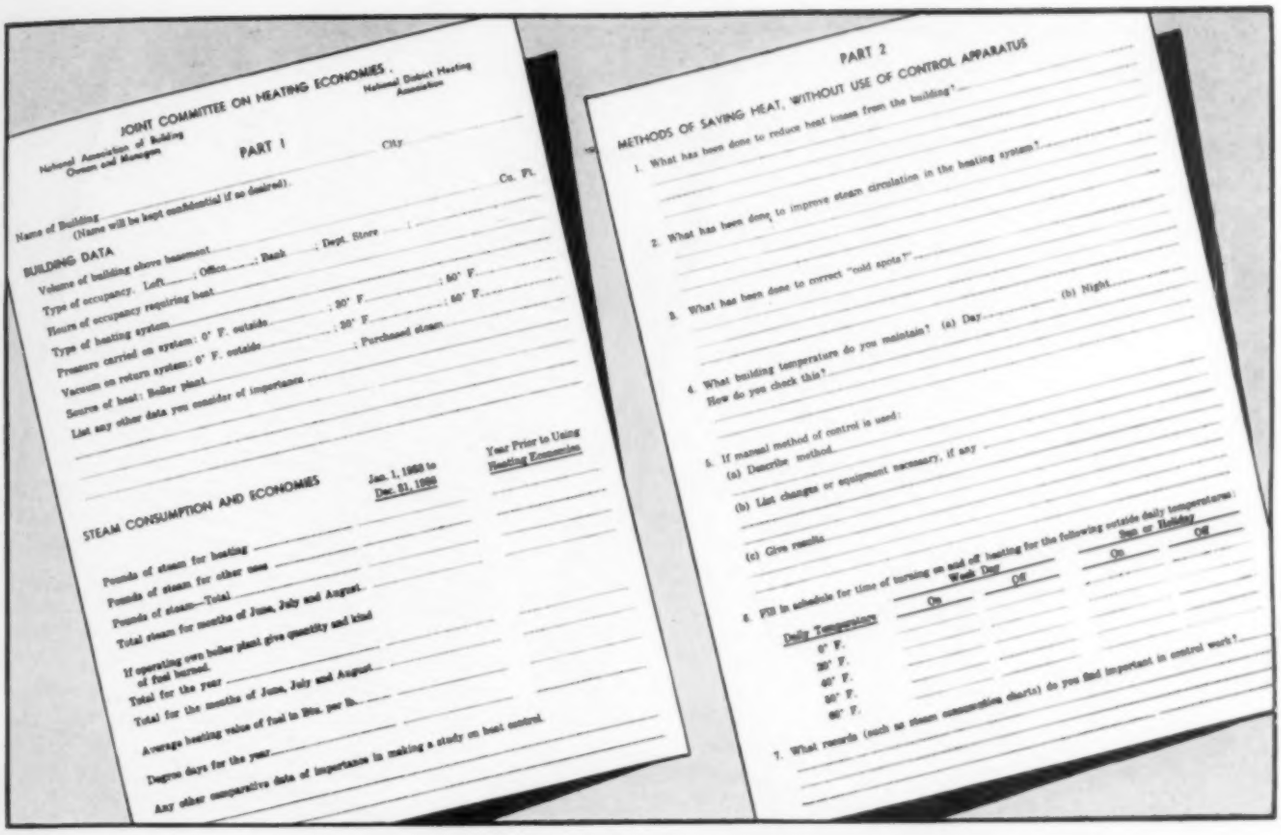
Table 2 groups the returns by cities. Nineteen cities reported and for about half of these the number of replies received was too small to permit safe conclusions. There has been some doubt as to whether there could be developed any accurate method of comparing the steam consumption of individual buildings or of buildings in different cities. This is based upon two things: (1) differences in construction and occupancy of the buildings themselves; and (2) differences in climatic conditions other than degree day data as between cities. While each individual case will vary somewhat from any standard, the committee believes the unit of lb of steam per degree day per 1000 cu ft of building volume can be used as a reasonable basis of comparison. If the average steam consumption for the several cities is compared with the average for the entire group, it will give at least an approximate measure of the efficiency with which buildings in each city are being operated. This is shown in the table under the heading "City Efficiency." It is interesting to note that St. Louis, having a normal of 4568 degree days per year, uses 121 per cent of the average amount of steam per degree day per 1000 cu ft; whereas Detroit, having 6415 degree days per year, takes only 88 per cent of the average amount of steam. In like manner, Portland, with 4447 degree days per year, uses 121 per cent of the average amount of steam; whereas Seattle, with almost the same number of degree days (4592), uses only 68 per cent of the average amount of steam. It is possible that in Detroit, Baltimore, New York, Philadelphia and Pittsburgh, all of which have low steam consumptions, the general use of purchased steam, supplemented by the intensive utilization work of the local steam companies, is a contributing factor.

Analysis by Source of Steam Supply

In the second study, Table 3, the replies were grouped according to the source of steam supply. One hundred and twenty-four buildings purchase steam; 29 buildings operate their own boiler plants. The buildings purchasing steam made a saving of 26.4 per cent, whereas those operating their own plants made a saving of 18.9 per

Table 3—Reports Grouped by Source of Steam Supply

Source of Steam	No. of Bldgs.	Volume 1000 Cu Ft	Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	Reports
			Before	After	Before	After		
Purchased	68	216,167	161,633	118,944	0.748	0.550	26.4	Complete
	56	135,323		80,375		0.594		Partial
	124	351,490		199,319		0.567		Total
Produced	11	45,721	40,635	32,977	0.889	0.721	18.9	Complete
	18	72,923		52,171		0.715		Partial
	29	118,644		85,148		0.718		Total



JOINT COMMITTEE ON HEATING ECONOMIES
National Association of Building Owners and Managers
National District Heating Association

PART 1

Name of Building (Name will be kept confidential if so desired) City

BUILDING DATA

Volume of building above basement Cu. Ft.
Type of occupancy. Loft ; Office ; Bank ; Dept. Store ;
Hours of occupancy requiring heat
Type of heating system
Pressure carried on system: 0° F. outside ; 20° F. ; 60° F.
Vacuum on return system: 0° F. outside ; 20° F. ; 60° F.
Source of heat: Boiler plant ; Purchased steam
List any other data you consider of importance

STEAM CONSUMPTION AND ECONOMIES

	Jan. 1, 1938 to Dec. 31, 1938	Year Prior to Using Heating Economies
Pounds of steam for heating		
Pounds of steam for other uses		
Pounds of steam—Total		
Total steam for months of June, July and August		
If operating own boiler plant give quantity and kind of fuel burned.		
Total for the year		
Total for the months of June, July and August		
Average heating value of fuel in Btu. per lb.		
Degree days for the year		

Any other comparative data of importance in making a study on heat control.

PART 2

METHODS OF SAVING HEAT, WITHOUT USE OF CONTROL APPARATUS

1. What has been done to reduce heat losses from the building?
2. What has been done to improve steam circulation in the heating system?
3. What has been done to correct "cold spots?"
4. What building temperature do you maintain? (a) Day (b) Night
   How do you check this?
5. If manual method of control is used:
   (a) Describe method
   (b) List changes or equipment necessary, if any
   (c) Give results
6. Fill in schedule for time of turning on and off heating for the following outside daily temperatures:

Daily Temperatures	Week Day		Sun or Holiday	
	On	Off	On	Off
0° F.				
20° F.				
40° F.				
50° F.				
60° F.				

7. What records (such as steam consumption charts) do you find important in control work?

The first and second pages of the questionnaire

cent. The buildings purchasing steam had a consumption rate, before economies, of 0.748 as against 0.889 for those buildings generating steam. Also, the buildings purchasing steam have a present consumption rate of 0.567 as against 0.718 for those buildings generating steam. It would thus appear that buildings purchasing steam consume 15.9 per cent less steam before control, and 21 per cent less steam after control, than those buildings operating their own plants. This is mainly due to two causes: (1) the elimination of boiler and plant losses inherent in plant operation; and (2) the closer regulation made possible with purchased steam, which can be turned on and off as desired.

It should be pointed out that this comparison would be more conclusive if there were more reports from buildings producing their own steam. The larger number of reports from buildings purchasing steam is probably due to the fact that such buildings have accurate records of their steam consumption, whereas those buildings producing their own steam either do not have accurate records or use coal figures rather than steam figures. With the amount of possible savings involved, the importance of having steam meters from which accurate records can be maintained is apparent.

Table 4—Reports Grouped by Type of Building Occupancy

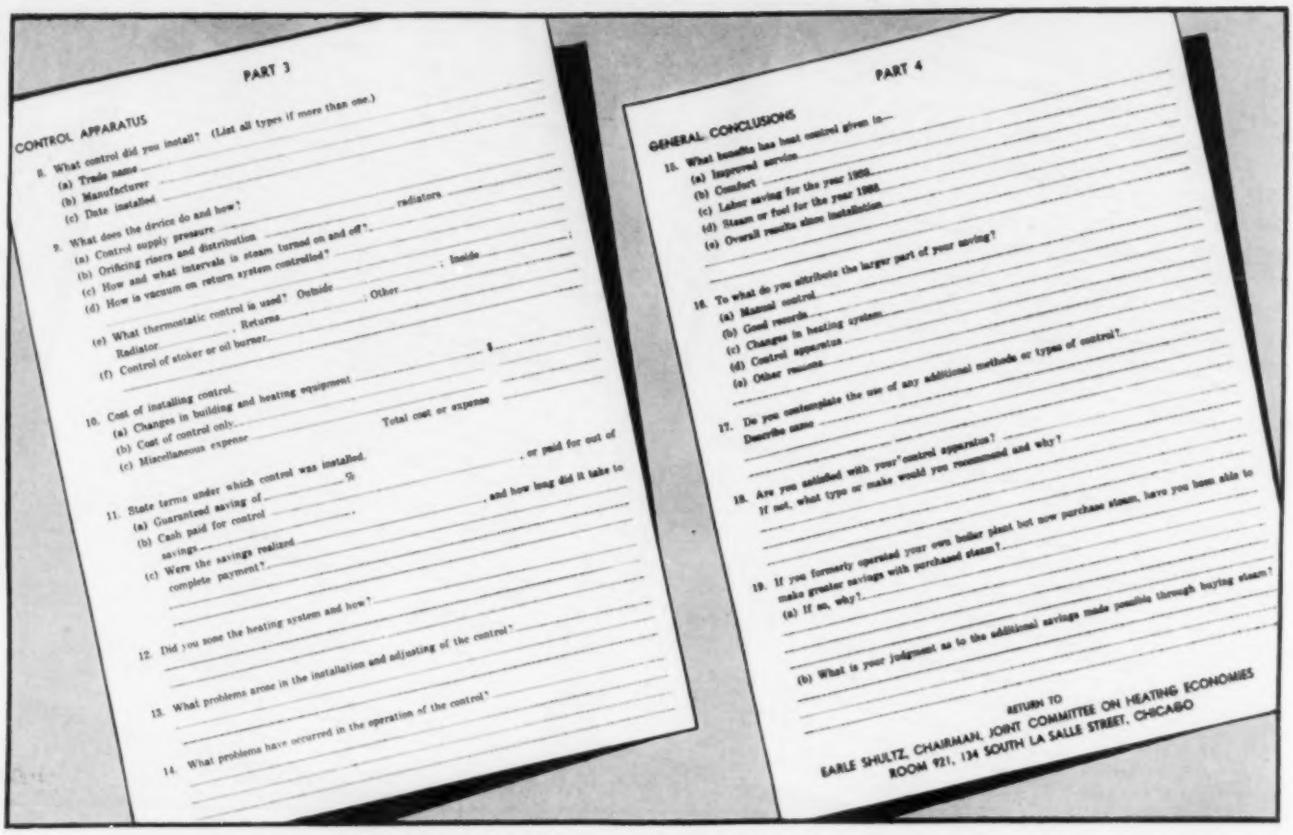
Type of Bldg.	No. of Bldgs.	Volume 1000 Cu Ft	Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	Reports
			Before	After	Before	After		
Office	54	187,902	148,468	111,478	0.790	0.593	25.0	Complete
	54	136,929		95,113		0.695		Partial
	108	324,831		206,591		0.636		Total
Office & Bank	7	38,815	27,880	19,430	0.718	0.501	30.3	Complete
	13	49,430		29,345		0.594		Partial
	20	88,245		48,775		0.552		Total
Bank	1	166	185	121	1.114	0.729	34.6	Complete
Total Office & Bank	62	226,883	176,533	131,029	0.777	0.578	25.7	Complete
	67	186,359		124,458		0.668		Partial
	129	413,242		255,487		0.619		Total
Stores	7	15,607	7,128	5,840	0.457	0.374	18.2	Complete
	2	12,392		2,860		0.231		Partial
	9	27,999		8,700		0.311		Total
Loft	5	11,122	6,750	5,782	0.607	0.520	14.3	Complete
	1	924		310		0.335		Partial
	6	12,046		6,092		0.506		Total
Hotels & Apartments	5	8,276	11,857	9,270	1.433	1.120	21.8	Complete
Theater	1	2,514		690		0.274		Partial
Garage	1	3,090		290		0.094		Partial
Misc.	2	2,967		3,938		1.327		Partial

Analysis by Type of Occupancy

In the third study, Table 4, the reports are grouped by type of building occupancy. Inasmuch as 84 per cent of the buildings reporting were office, or office and bank, buildings the statistics for the remaining classifications are inadequate for definite conclusions. They are, however, fairly in line with the figures used by heating companies in estimating these classes of business.

Analysis by Type of Heating

In the fourth study, Table 5, the reports are grouped by types of heating systems. Here again the limited number of questionnaires makes accurate conclusions difficult. There are two classes, however, with a sufficient number of returns to justify conclusions. They are the buildings having a two



PART 3

CONTROL APPARATUS

8. What control did you install? (List all types if more than one.)
(a) Trade name
(b) Manufacturer
(c) Date installed

9. What does the device do and how?
(a) Control supply pressure
(b) Orificing risers and distribution
(c) How and what intervals is steam turned on and off?
(d) How is vacuum on return system controlled?
(e) What thermostatic control is used? Outside ; Inside ; Radiator ; Returns ; Other
(f) Control of stoker or oil burner

10. Cost of installing control.
(a) Changes in building and heating equipment $
(b) Cost of control only
(c) Miscellaneous expense
Total cost or expense

11. State terms under which control was installed.
(a) Guaranteed saving of %
(b) Cash paid for control , or paid for out of savings
(c) Were the savings realized , and how long did it take to complete payment?

12. Did you zone the heating system and how?

13. What problems arose in the installation and adjusting of the control?

14. What problems have occurred in the operation of the control?

PART 4

GENERAL CONCLUSIONS

15. What benefits has heat control given in—
(a) Improved service
(b) Comfort
(c) Labor saving for the year 1938
(d) Steam or fuel for the year 1938
(e) Overall results since installation

16. To what do you attribute the larger part of your saving?
(a) Manual control
(b) Good records
(c) Changes in heating system
(d) Control apparatus
(e) Other reasons

17. Do you contemplate the use of any additional methods or types of control? Describe same

18. Are you satisfied with your control apparatus? If not, what type or make would you recommend and why?

19. If you formerly operated your own boiler plant but now purchase steam, have you been able to make greater savings with purchased steam?
(a) If so, why?
(b) What is your judgment as to the additional savings made possible through buying steam?

RETURN TO
EARLE SHULTZ, CHAIRMAN, JOINT COMMITTEE ON HEATING ECONOMIES
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Third and fourth pages of the questionnaire form

pipe vacuum system and the buildings using orifices in the radiators and heating systems. The first class has 66 buildings and the second class has 33 buildings. Both of these classes use less than the average amount of steam, whereas all of the other types (with the exception of the one pipe vacuum system, where only three buildings reported) use considerably more than the average amount of steam. As between the two systems, the two pipe vacuum system used an average of 0.581 lb of steam per degree day per 1000 cu ft of volume, whereas the orificed system used 0.546 lb of steam, or a saving of 6 per cent. It would seem, therefore, that there is a definite advantage in the use of orifices.

[The rest of the survey is to appear next month, and will analyze the reports by type of control, size of buildings, hours of heating, and cost of control.]

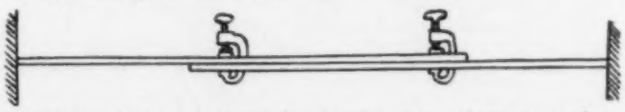
Table 5—Reports Grouped by Type of Heating System

Htg. Sys.	No. of Bldgs.	Volume 1000 Cu Ft	Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	Reports
			Before	After	Before	After		
1 Pipe Grav.	13	6,703	6,219	5,018	0.928	0.749	19.3	Complete
	10	16,160		13,063		0.808		Partial
	23	22,863		18,081		0.791		Total
2 Pipe Grav.	7	8,036	10,312	7,729	1.283	0.962	25.0	Complete
	6	11,839		8,077		0.682		Partial
	13	19,875		15,806		0.795		Total
Air Line (Paul)	6	21,321	23,510	15,830	1.103	0.742	32.8	Complete
	4	7,688		6,015		0.782		Partial
	10	29,009		21,845		0.753		Total
1 Pipe Vac.	3	6,278		4,160		0.663		Partial
2 Pipe Vac.	27	123,899	97,544	71,083	0.787	0.574	27.2	Complete
	39	146,930		86,173		0.587		Partial
	66	270,829		157,256		0.581		Total
Vapor (Pump or Grav. Return)	1	49	100	60	2.041	1.224	40.0	Complete
	4	2,300		2,290		0.996		Partial
	5	2,349		2,350		1.000		Total
Orificed Grav.	11	39,219	25,968	20,951	0.662	0.534	19.3	Complete
	4	7,608		5,400		0.710		Partial
	15	46,827		26,351		0.563		Total
Orificed Vac.	14	62,661	38,615	31,260	0.616	0.499	19.0	Complete
	4	9,443		7,368		0.780		Partial
	18	72,104		38,628		0.536		Total
Total Orificed	25	101,880	64,583	52,211	0.636	0.513	19.3	Complete
	8	17,051		12,768		0.749		Partial
	33	118,931		64,979		0.546		Total

A Handy Measuring Kink

A measurement that is difficult to take accurately is the exact distance between two walls or objects which cannot be measured directly with a rule, cannot be measured with a stick that is too long, or with one that is too short.

An excellent kink for obtaining the distance is shown in the accompanying sketch. Use two sticks such as laths or anything else sufficiently stiff. The total length of the two must be somewhat greater than the distance between the walls or objects being measured so that they will overlap as shown. Put them together in contact as indicated and slide outward until the two ends make contact with the desired points. Then clamp or fasten them together as shown, and you have the exact distance.



This method is decidedly better than the cut and try method which I used before I became acquainted with this time- and temper-saving kink.—W. F. Schaphorst.

The Psychrometric Chart

Its Application and Theory

By William Goodman*

FOR those who desire a convenient method of locating the wet bulb lines, the values in Tables 2 to 13 can be used. The end points for the normal barometric pressure of 29.92 in. of mercury are given in Table 2. In Tables 3 to 13 inclusive the end points are given for barometric pressures from 22 to 32 in. inclusive. These end points show where a given line of constant wet bulb temperature intersects either the line of constant dry bulb temperature or the line of constant absolute humidity which forms the border of each chart. For example, on the middle range psychrometric chart (Chart A), the highest dry bulb temperature shown is 120 deg. If the end point for a wet bulb temperature of 72 deg is to be found for a barometric pressure of 29.92 in., refer to the column headed ($t = 120$ deg) in Table 2.† The end point is given as $w = 0.005851$ lb. This means that the line of 72 deg wet bulb temperature crosses the dry bulb temperature line of 120 deg at the same point where the horizontal line of $w = 0.005851$ lb of absolute humidity crosses. Point 1 of Fig. 12 was located in this way. Point 2 of this same figure is located where the line of 72 deg dry bulb temperature crosses the saturation curve for a barometric pressure of 29.92 in. The line of constant wet bulb temperature is then drawn between points 1 and 2. Similarly, values of dry bulb temperature along lines of constant absolute humidity are given in these tables. For example, in the column headed $w = 0$, the end point is given as 83.91 deg for a wet bulb temperature of 50 deg. This end point is located along $w = 0$ at point 3 of Fig. 12. Point 4 is located where the line 50 deg dry bulb temperature crosses the saturation curve. The wet bulb temperature line for 50 deg is then drawn through points 3 and 4. For wet bulb temperature lines which do not cross the saturation curve on the psychrometric chart, two end points are given in the tables so that the wet bulb line can be drawn between them.

By using the end points and the tables for the various barometric pressures, lines of constant wet bulb temperature can be drawn for any barometric pressure. Lines of constant wet bulb temperature can be drawn more conveniently and accurately by means of these tables of end points than is possible by using ratio angles.

All of the end points in Tables 2 to 13 were computed by equation 25, which is presented later.

Mixtures of Air and Steam

As previously stated, equations 3 and 5, and consequently equation 12, are equally applicable if moisture in the form of steam is added to the air instead of in the form of liquid. In either case, h_o represents the initial enthalpy of the moisture regardless of whether it is in the form of steam or water.

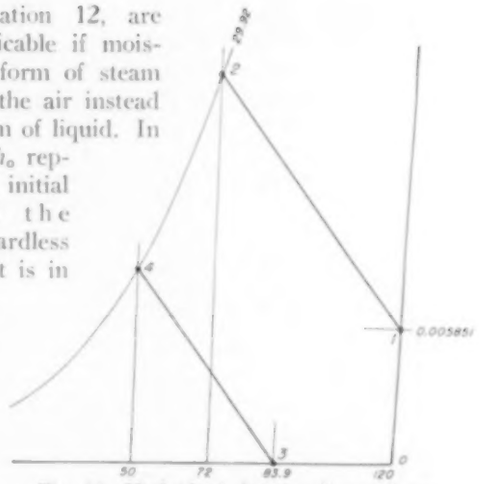


Fig. 12—Method of drawing lines of constant temperature of adiabatic saturation on psychrometric chart by means of tables of end points

The enthalpy of saturated steam at 2 lb gage is 1153 Btu. Referring to Table 1, for this enthalpy the ratio angle is about 86°. Line 1-3 in Fig. 13 has been drawn for this angle. Evidently, if saturated steam at this temperature is added, little change will take place in the dry bulb temperature of the air. The moisture that is being added to the air has already been evaporated; the latent heat required for this evaporation has been acquired elsewhere. Hence, this case represents nothing more than the mixing of two gases—air and steam.

The flow of high pressure steam through a nozzle into the free atmosphere is known in thermodynamics as a throttling process. The final enthalpy of the steam sprayed into the air is equal to its initial enthalpy. In addition, the temperature of the steam as it leaves the nozzle and is sprayed into the air is about the same as the temperature of the high pressure steam ahead of the nozzle.

In addition to the elementary uses of the psychrometric chart—such as showing the relationships between dry bulb, wet bulb, and dew point temperatures, and relative humidity—it provides a simple and enlightening means of analyzing many complex problems in comfort and process air conditioning, including drying. . . . New psychrometric charts covering the low, middle and high temperature ranges, with saturation curves for various barometric pressures, are presented here. The general theory of the psychrometric chart is discussed in detail, and the methods of solving problems by the use of charts are described. . . . These new charts facilitate solution of most problems and lead to clearer understanding of the fundamentals

*The Trane Co. Member of Board of Consulting and Contributing Editors.

†See table of symbols, p. 358, June issue.

Part 5. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, June, 1939, pp. 357-360; Part 2 in July, pp. 421-424; Part 3 in August, pp. 485-487; and Part 4 in September, pp. 549-551.

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Table 2 - End Points for Wet Bulb Lines
Barometric Pressure 29.921 In.

Wet Bulb Temp t'	Dry Bulb Temperature, t		Absolute Humidity, w					
	For w = 0		For w = 0.022	For w = 0.11	For t = 40		For t = 120	For t = 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
-20	-18.23	-18.40						
-19	-17.14	-17.31						
-18	-16.04	-16.23						
-17	-14.94	-15.13						
-16	-13.83	-14.04						
-15	-12.71	-12.94						
-14	-11.60	-11.83						
-13	-10.47	-10.72						
-12	-9.34	-9.60						
-11	-8.21	-8.48						
-10	-7.06	-7.36						
-9	-5.92	-6.23						
-8	-4.75	-5.09						
-7	-3.60	-3.94						
-6	-2.43	-2.79						
-5	-1.25	-1.63						
-4	-0.07	-0.47						
-3	+1.12	+0.70						
-2	+2.33	+1.88						
-1	+3.54	+3.07						
0	4.75	4.26						
1	5.98	5.46						
2	7.21	6.67						
3	8.46	7.89						
4	9.73	9.12						
5	10.99	10.36						
6	12.27	11.60						
7	13.56	12.86						
8	14.86	14.13						
9	16.18	15.41						
10	17.51	16.71						
11	18.86	18.01						
12	20.22	19.32						
13	21.59	20.64						
14	22.98	21.99						
15	24.38	23.34						
16	25.79	24.71						
17	27.23	26.09						
18	28.68	27.49						
19	30.15	28.90						
20	31.63	30.32						
21	33.15	31.77						
22	34.67	33.23						
23	36.22	34.71						
24	37.79	36.21						
25	39.36	37.72						
26	40.98	39.26			0.0001927			
27	42.61	40.81			0.0005118			
28	44.27	42.38			0.0008367			
29	45.95	43.98			0.0011167			
30	47.66	45.60			0.0015041			
31	49.41	47.25			0.0018461			
32	51.16	48.90			0.0022031	0.0019933		
33		50.61				0.0023611		
34		52.32				0.0027461		
35		54.07				0.0031377		
36		55.84				0.0035536		
37		57.64				0.0039941		
38		59.46				0.0043571		
39		61.31				0.0047721		
40		63.19						
41		65.10						
42		67.05						
43		69.03						
44		71.03						
45		73.10						
46		75.18						
47		77.30						
48		79.45						
49		81.65						
50		83.91						
51		86.18						
52		88.50						
53		90.88						
54		93.29						
55		95.74						
56		98.26						
57		100.8						
58		103.4						
59		106.1						
60		108.8						
61		111.5						
62		114.4						
63		117.3						
64		120.2					0.0000508	

Wet Bulb Temp t'	Dry Bulb Temperature, t		Absolute Humidity, w					
	For w = 0		For w = 0.022	For w = 0.11	For t = 40		For t = 120	For t = 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
65		123.3					0.0007255	
66		126.3					0.001401	
67		129.5					0.002097	
68		132.7					0.002814	
69		136.0					0.003549	
70		139.3					0.004295	
71		142.7					0.005064	
72		146.2					0.005851	
73		149.8					0.006658	
74		153.5					0.007488	
75		157.2					0.008327	
76		161.1					0.009195	
77		165.0					0.01009	
78		169.0					0.01099	
79		173.0					0.01192	
80		177.2	81.05				0.01287	
81		181.5	85.28				0.01385	
82		185.9	89.59				0.01486	
83		190.4	94.02				0.01589	
84		195.0	98.49				0.01693	
85		199.7	103.1				0.01802	
86		204.6	107.9				0.01913	
87		209.5	112.7				0.02027	
88		214.6	117.6				0.02143	
89		219.7						
90		225.1						
91		230.5						
92		236.1						
93		241.8						
94		247.7						
95		253.7						
96		259.8						
97		266.1						
98		272.6						
99		279.2						
100		286.0						
101		293.0						
102		300.2						
103		307.6						
104		315.0						
105		322.8						
106		330.7						
107		338.9						
108		347.2						
109		355.9						
110		364.6						
111		373.7						
112		383.0						
113		392.6						
114		402.4						
115		412.5						
116		422.9						
117		433.5						
118		444.4						
119								0.001140
120								0.003514
121								0.005969
122								0.008466
123								0.01107
124								0.01369
125								0.01647
126								0.01930
127								0.02226
128								0.02523
129								0.02830
130				134.6				0.03155
131				148.3				0.03488
132				161.9				0.03822
133				176.6				0.04182
134				191.3				0.04542
135				206.3				0.04912
136				222.4				0.05308
137				238.8				0.05713
138				255.5				0.06128
139				272.5				0.06551
140				290.6				0.07000
141				308.7				0.07451
142				328.1				0.07936
143				347.2				0.08413
144				367.7				0.08926
145				388.5				0.09448
146				410.6				0.1001
147				433.1				0.1057

Table 3 - End Points for Wet Bulb Lines
Barometric Pressure 22 In.

Wet Bulb Temp t'	Dry Bulb Temperature, t		Absolute Humidity, w					
	For w = 0		For w = 0.022	For w = 0.11	For t = 40		For t = 120	For t = 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
-20	-17.59	-17.82						
-19	-16.46	-16.71						
-18	-15.33	-15.59						
-17	-14.19	-14.46						
-16	-13.04	-13.33						
-15	-11.89	-12.19						
-14	-10.73	-11.05						
-13	-9.56	-9.90						
-12	-8.38	-8.74						
-11	-7.20	-7.58						
-10	-6.00	-6.40						
-9	-4.80	-5.22						
-8	-3.59	-4.03						
-7	-2.37	-2.84						
-6	-1.14	-1.64						
-5	+0.10	-0.42						
-4	1.35	+0.80						
-3	2.61	2.04						
-2	3.89	3.28						
-1	5.17	4.54						
0	6.46	5.79						
1	7.77	7.07						
2	9.10	8.36						
3	10.43	9.65						
4	11.79	10.97						
5	13.16	12.29						
6	14.54	13.63						
7	15.93	14.98						
8	17.35	16.35						
9	18.78	17.73						
10	20.23	19.12						
11	21.70	20.54						
12	23.19	21.97						
13	24.68	23.40						
14	26.22	24.88						
15	27.77	26.36						
16	29.34	27.86						
17	30.92	29.38						
18	32.54	30.92						
19	34.18	32.48						
20	35.85	34.07						
21	37.54	35.68						
22	39.26	37.30						
23	41.00	38.95			0.0001947			
24	42.78	40.63			0.0005436	0.0001398		
25	44.57	42.32			0.0008938	0.0005133		
26	46.42	44.07			0.001258	0.0009009		
27	48.28	45.82			0.001623	0.001289		
28	50.17	47.60			0.001993	0.001694		
29	52.10	49.41			0.002373	0.002097		
30	54.07	51.26			0.002761	0.002500		
31	56.09	53.15			0.003158	0.002925		
32	58.11	55.05			0.003555	0.003345		
33		57.00				0.003784		
34		58.98				0.004228		
35		60.99				0.004680		
36		63.05				0.005144		
37		65.14				0.005617		
38		67.27				0.006098		
39		69.44				0.006588		
40		71.65				0.007090		
41		73.89						
42		76.19						
43		78.52						
44		80.90						
45		83.35						
46		85.83						
47		88.36						
48		90.96						
49		93.60						
50		96.31						
51		99.10						
52		101.9						
53		104.8						
54		107.7						
55		110.7						
56		113.8						
57		116.9						
58		120.2					0.00003489	
59		123.4					0.0007555	

Wet Bulb Temp t'	Dry Bulb Temperature, t		Absolute Humidity, w					
	For w = 0		For w = 0.022	For w = 0.11	For t = 40		For t = 120	For t = 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
60		126.8					0.001496	
61		130.2					0.002258	
62		133.8					0.003048	
63		137.4					0.003849	
64		141.0					0.004661	
65		144.9					0.005514	
66		148.7					0.006367	
67		152.6					0.007251	
68		156.7					0.008185	
69		160.9					0.009099	
70		165.1					0.01005	
71		169.5	72.77				0.01103	
72		173.9	77.18				0.01205	
73		178.5	81.68				0.01308	
74		183.2	86.29				0.01415	
75		188.0	90.95				0.01523	
76		192.9	95.76				0.01634	
77		198.1	100.8				0.01751	
78		203.2	105.9				0.01869	
79		208.5	111.0				0.01990	
80		214.0	116.3				0.02114	
81		219.6						
82		225.3						
83		231.2						
84		237.2						
85		243.4						
86		249.8						
87		256.4						
88		263.0						
89		270.0						
90		277.1						
91		284.4						
92		291.8						
93		299.5						
94		307.4						
95		315.5						
96		323.7						
97		332.2						
98		340.7						
99		349.2						
100		359.4						
101		369.0						
102		378.9						
103		389.0						
104		399.4						
105		410.0						
106		421.1						
107		432.5						
108		444.2						
109								0.001258
110								0.003779
111								0.006347
112								0.009030
113								0.01180
114								0.01465
115								0.01758
116								0.02061
117								0.02376
118								0.02698
119							126.4	0.03030
120							140.9	0.03380
121							155.7	0.03738
122							170.8	0.04106
123							186.7	0.04490
124							200.8	0.04885
125							218.7	0.05297
126							236.9	0.05718
127							254.8	0.06157
128							273.3	0.06613
129							292.6	0.07088
130							312.5	0.07581
131							333.1	0.08091
132							353.7	0.08601
133							375.7	0.09147
134							398.7	0.09719
135							422.2	0.1030
136							446.4	0.1091

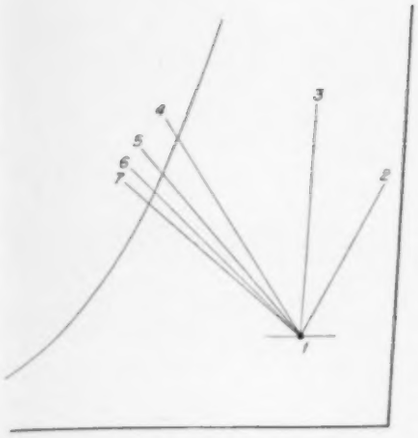


Fig. 13—Relative location of ratio lines on psychrometric chart for moisture in various forms. The angles in this illustration have been exaggerated for the sake of clarity

Ratio line *1-2* corresponds to highly superheated steam.
Ratio line *1-3* corresponds to saturated steam.
Ratio line *1-4* corresponds to water at 212 degrees.
Ratio line *1-5* corresponds to water at 32 degrees.
Ratio line *1-6* corresponds to ice at 32 degrees.
Ratio line *1-7* corresponds to ice at —40 degrees.
Any ratio line for superheated steam will fall in the sector between lines *1-2* and *1-3*. The ratio line corresponding to wet steam, that is to a mixture of steam and water, will fall in the sector between lines *1-3* and *1-4*. The ratio line for water at any temperature will fall in the sector between lines *1-4* and *1-5*. The ratio line corresponding to a mixture of ice and water will fall in the sector between lines *1-5* and *1-6*. The ratio line for ice at any temperature between 32 deg and —40 deg will fall in the sector between lines *1-6* and *1-7*.

When steam is sprayed into air and the two are thoroughly mixed, the temperature of the steam will fall and the temperature of the air will rise until the temperature of the two is the same. During the mixing process, the steam surrenders the heat that raises the temperature of the air. Because the weight of steam added to the air is very small, it can surrender only a little heat. Consequently, the increase in the temperature of the air is very small as long as the air remains unsaturated. The case in which the air becomes saturated and some of the steam condenses is discussed later.

The moisture ratio which determines the slope of any line of dry bulb temperature on the psychrometric chart is, as shown in Appendix 4,

$$q_D = h_g \quad \text{.....[15]}$$

$$q_D = 1061 + 0.45t \quad \text{.....[16]}$$

where q_D = moisture ratio or slope of any given line of dry bulb temperature;

h_g = enthalpy of saturated steam at the given dry bulb temperature, Btu per lb;

t = dry bulb temperature of the line whose slope is q_D.

The moisture ratio which determines the slope of the ratio line when steam is added to air is given by $q = h_o$, *where* h_o = enthalpy of the steam added to the air.

The meanings of h_o and h_g should not be confused; the two are different: h_o is the initial enthalpy of the steam before it is mixed with the air, whereas h_g is the enthalpy of the steam mixed with the air and is at the same temperature as the dry bulb temperature of the air.

If q is greater than q_D, the dry bulb temperature of the air will increase as the steam is added to the air. q can be greater than q_D only if h_o, the initial enthalpy of the steam, is greater than h_g, the enthalpy of the vapor at a temperature equal to the initial dry bulb temperature of the air. If $h_o = h_g$, then there will be no increase in the dry bulb temperature of the air. As shown in the following paragraph, for most ordinary conditions h_o is greater than h_g because the temperature of the steam sprayed into the air is greater than the temperature of the air. Nevertheless, because the weight of steam added to the air is usually very small little heat is surrendered by the steam; the increase in the temperature of the air is accordingly small if the air remains unsaturated.

In order to spray steam into air, the pressure of the steam ahead of the nozzle must be greater than the barometric pressure of the air. For air at ordinary barometric pressure, the steam pressure must be greater than 14.7 lb per sq in. Now the temperature and enthalpy of saturated steam at this pressure are 212 deg and 1150.4 Btu, respectively. Consequently, if steam is sprayed into air whose dry bulb temperature is less than 212 deg, the temperature of the air must increase slightly—the increase depending upon the weight of steam mixed with the air.

If saturated steam initially at 14.7 lb per sq in. is sprayed into air whose dry bulb temperature is greater than 212 deg, the dry bulb temperature of the air will fall slightly because the temperature of the steam being sprayed into the air is lower than the dry bulb temperature of the air. Hence, the air must cool slightly as it surrenders a small amount of heat to raise the temperature of the superheated steam. Whether the air will be heated or cooled by the addition of steam depends on whether the initial enthalpy of the steam is higher or lower than the enthalpy of the vapor initially mixed with the air.

If, instead of saturated steam, superheated steam is added to air, the ratio line will incline to the right. The maximum superheat tabulated in the steam tables is 1600 deg. For this degree of superheat, the maximum enthalpy is slightly less than 1860 Btu. From Table 1 for $q = 1860$, the ratio angle is 60°-48′. For steam superheated 1600 deg, the final condition of the air leaving the conditioning chamber would be represented by a point along the line *1-2* of Fig. 13. The ratio lines for steam at various degrees of superheat will lie somewhere between lines *1-2* and *1-3*.

In equation 12, h_o can also represent the enthalpy of moisture in the form of wet steam, that is, steam having a quality less than one. If steam at a pressure of 2 lb gage has a quality of 50 per cent (that is, it consists of 50 per cent steam and 50 per cent water carried in suspension in the steam) its enthalpy, h_o, as determined by conventional methods would be 669.7 Btu (186.6 + 0.5 × 966.1 = 669.7 Btu). Using Table 1 for q = 669.7 Btu, the ratio angle is 74°-42′. Referring to Fig. 13, the ratio line for wet steam must evidently lie somewhere between the line *1-4* for liquid at 212 deg and the line *1-3* for saturated steam. From this, if wet steam is sprayed into air, the dry bulb temperature of the air must drop. The reason for this is that the air cools in providing the heat necessary to evaporate the liquid mixed with the steam.

In Fig. 13, line *1-4* for liquid at 212 deg and line *1-5* for liquid at 32 deg have the same slope as the corresponding lines shown in Fig. 8. The ratio lines for liquid at any temperature must, therefore, lie between the lines *1-4* and *1-5*. In the same way, in Fig. 13, line *1-6* corresponds to the enthalpy of ice at 32 deg and line *1-7*

[*Continued on page* 617]

It Pays to Keep Cool

Detroit Edison's drafting and surveying bureau

By **Paul G. Heidman***

The Detroit Edison Co. has measured the benefits of office air conditioning in its new service building, finds that it increases efficiency over 50 per cent, reduces operating and maintenance expenses more than 39 per cent. . . . Because of air conditioning, an economical, solid block building design was made possible, so that this six story structure contains as much usable space as would a conventional eight story one

UNTIL recently, few figures were available by which the dollar-and-cents value of air conditioning an office could be measured. With a view to finding out more about it, a study was made in the drafting and surveying bureau of The Detroit Edison Co. over the past two years. A definite unit of work was chosen for the test—one which would serve as a fair indicator—and figures were kept during June, July and August of 1937 and 1938. During the summer of 1937, in offices *without* air conditioning, 8988 work units required 5008 man-hours. During the summer of 1938, in the new air conditioned service building, 10,474 work units required only 3872 man-hours—an increase in efficiency of 51.4 per cent.

An interesting comment was made by M. E. Sickelsteel of the drafting and surveying bureau. Pointing out that air conditioning contributes not only to comfort, but also to health, he said: "Several times last summer, an employee came back to work sooner after a sick leave because he was *more at ease working in our quarters* than he was resting at home." The salutary effect of air conditioned surroundings on hay fever victims is well known. People in an air conditioned office breathe clean air—free from dust and pollen. Those having hay fever and asthma frequently obtain a large measure of relief.

Mr. Sickelsteel also commented on the effect of air conditioning in connection with a technical aspect of the work in the map divison. The tracing of maps from original drawings formerly was complicated by the stretching of tracing cloth on damp days. This cloth is made of a specially treated cotton fabric, and the fibers stretch appreciably under humid conditions. When sections of a large tracing were made on two or three successive days, they would not fit if placed side by side, although they had been drawn from adjoining parts of the same map. Humidity control has solved this annoying problem.

Considering the many advantages obtained through air conditioning, one might naturally ask how much it cost to incorporate it in the new service building. And one discovers the altogether amazing answer that it cost nothing at all! Or rather, that the air conditioning in the building paid for itself, which is perhaps a different

*Superintendent of Buildings, The Detroit Edison Co.

way of saying the same thing. How was this accomplished? The answer is a lesson in architectural design.

In the past, office buildings have been designed so that as much floor space as possible would be near a window, because the window was regarded as the essential source of light and ventilation. Consequently, narrow wings were necessary, with floor plans resembling the letters U, H, or E. In the Detroit Edison service building, the simple and economical solid block design was made possible by three things: (1) Complete year 'round air conditioning, (2) a unique system of artificial lighting, combined with glass block walls, and (3) a style of architecture that provides maximum usable space at minimum cost.[1]

There is nothing fancy or elaborate about this service building. It is a plain, practical structure—not expensive. But thanks to lighting and air conditioning, this six story building *with air conditioning* cost no more to build than, and contains as much usable space as, a conventional eight story building of the U, H, or E type *without* air conditioning.

Thanks to air conditioning, the total operating and maintenance costs for the new service building (based on net area and corrected for the difference in the size of building) are 39.7 per cent lower than those in the company's general office building, figured for the last six months of 1938. Labor costs are 40.2 per cent lower, steam costs 62.8 per cent lower, cleaning costs 42.7 per cent lower. Electricity costs are only 1.6 per cent higher although the light intensity is twice that in the general office building. Electricity costs include the power for all air conditioning equipment.

Refrigeration equipment for the service building

Air conditioning has been used for some time in manufacturing and industrial plants and in laboratories to create ideal conditions of "artificial weather." Temperature and humidity are factors which—if not carefully controlled—often affect processes and chemical reactions, as well as machine operations. A good example of this may be found in our own addressograph department. Tempering of the air has almost eliminated static —a bane of all printing devices. Formerly, static made cards pile up unevenly and stick together so tightly that they could not be evened up without stopping the operation of the machine. Also, cards stuck to the mechanism and buckled, then tore. Air conditioning has removed this obstacle.

Instances of improved operation in other departments are also worth noting. In the customers' accounts division, a good deal of work is done on Hollerith machines —sensitive equipment used in the company's system of billing. These machines perform almost the entire billing operation *automatically*—printing, figuring, balancing, checking and sorting bills mechanically—with a precision and accuracy that is little short of uncanny. But their operation revolves around an office record card perforated like a section of a player piano roll. In the various machines in which these key-punched cards are used, such fine adjustment is necessary that a wet day would often raise havoc with the cards. The machine is designed to pull cards off a pile one at a time, and push them through a very narrow space. In the moist atmosphere, the cards would tend to buckle—thereby clogging the machine and necessitating stopping it while the spoiled cards were removed. Air conditioning has successfully remedied this difficulty.

In the testing laboratories of the meter department, clean conditioned air means less dust on instruments to impair their accuracy. Air conditioning permits shutting off outside noises and distractions from the close work of the standards section. Variations in room temperatures are minimized, reducing changes in the current delivered by standard cells.

Of special interest in view of the air conditioning is a research study on the accumulation of dust in the new service building. Careful tests show that the rate of dust accumulation is *approximately one-fourth* that in the company's general office building. The dust in the service building is light colored and not as noticeable as the dark dust present in the general office building—so that cleaner working conditions prevail in the service building, which is dusted once a week, than in the general offices, which are dusted every night.

The service building furnishes a good example of the advantages and benefits to be derived from air conditioning. And it serves to illustrate a significant point: What has been done here can be done anywhere, in a small store, an office, a factory. The differences are chiefly differences in the size of equipment and details; the net result is the same. It is highly probable that some day year around comfort will not be the exception, but the rule.

[1]The design and air conditioning of the service building were described in detail in two articles published in HEATING, PIPING AND AIR CONDITIONING: Air Conditioning Influences Building Design, by J. H. Walker, October, 1937; and Unusual Problems Met in Designing New Glass Block Building, by James N. Livermore, April, 1938.

Psychrometric Chart—

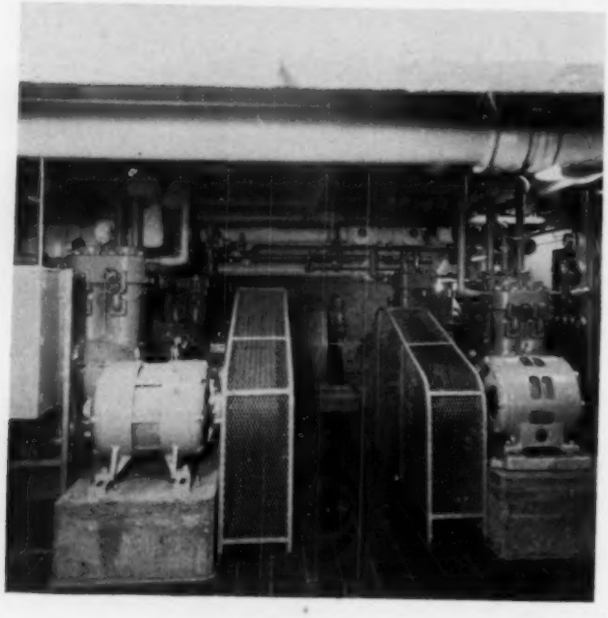
[*Continued from page* 615]

corresponds to the enthalpy of ice at —40 deg. These latter two lines will be discussed in more detail later. However, it is interesting to note that any ratio line corresponding to a mixture of ice and water must lie between lines *1-5* and *1-6;* the exact location depends upon the enthalpy of the mixture of ice and water, and this in turn depends upon the proportions of ice and water present in the mixture. Any ratio line corresponding to the enthalpy of ice having any temperature between 32 deg and —40 deg must lie between the lines *1-6* and *1-7*. Fig. 13 shows the complete field in which ratio lines for moisture in various states must lie.

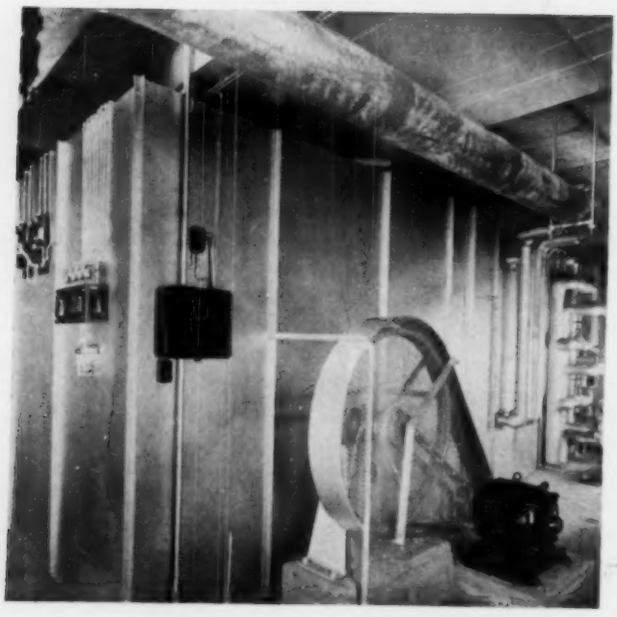
[*To be continued*]

Kraft Cheese Has New Air Conditioned Headquarters

Top left—The Kraft-Phenix Cheese Corporation's new world headquarters building, completed earlier this year in Chicago, represents one of the major building projects of the past 10 years. With many modern features—including air conditioning—it houses executive and general offices, research laboratories, and a factory producing several Kraft products for the midwest market. The central tower portion of the building is 10 stories high, and the manufacturing units five stories. Occupancy is approximately 900 people



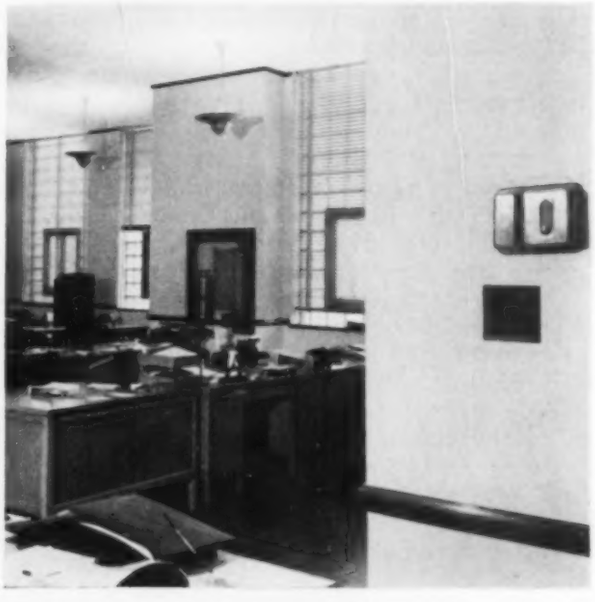
Center left—The ammonia compressor installation in the basement. Chilled water conveys refrigerating effect to the central station type air conditioning equipment on the top floor and to units for the factory offices, lunchrooms, etc. Thirty-four degree brine is circulated to the process conditioning units, and 15 deg brine and direct expansion systems serve ice boxes and general refrigeration requirements. Condensing water is cooled in a three section cooling tower on the roof, one section of which has a variable speed fan. The spray headers are split into two 800 gpm and one 300 gpm sections, each with separate pumps, so that the operation can be matched to the refrigeration load. When the load is light, the cooling tower operates as an atmospheric tower



Bottom left—One of the three central station air conditioning systems, each of which serves one of the upper office floors. Summer and winter combination thermostats and humidistats control the conditions maintained the year around for each air conditioned zone. During summer operation, the setting of the fan discharge thermostat for each system is automatically readjusted by a static pressure regulator. When the static pressure in the supply duct rises due to the various branch cut-off dampers gradually closing, a decrease in the cooling load is indicated and the setting of the fan discharge temperature thermostat is therefore automatically raised according to a predetermined schedule

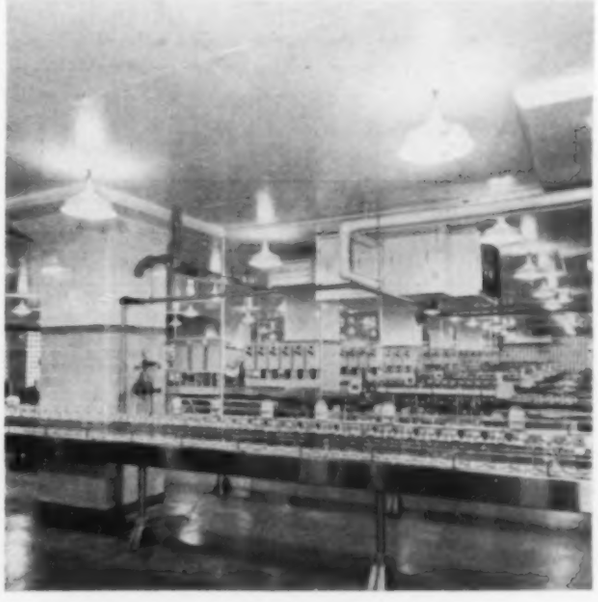
Top right—View of a general office area in the Kraft-Phenix building. The ceiling fixtures—designed through the coöperation of air conditioning and illuminating engineers—combine the air grilles and the lighting units and are spaced about 10 ft apart

Center right—One of the combination summer-winter thermostats and humidistats which control temperature and humidity throughout the year for each air conditioning zone



Bottom right—A view in one of the factory departments of the Kraft-Phenix building showing some of the suspended type air conditioning units which serve these areas, the excellent lighting, and the efficient arrangement of the plant's equipment. Some of the conditioning units recirculate all the air they handle, and others have outside air connections, depending upon the ventilation requirements . . . . Wrought iron pipe is used for chilled water lines, brine lines, steam returns, and concealed hot and cold water lines in this building and stainless and seamless steel tubing for conveying mayonnaise and salad dressing. Unit heaters are used in manufacturing spaces and convectors care for wall heat losses. Two 300 hp oil fired boilers operating at 125 lb pressure furnish the steam requirements . . . . The architects and engineers were Mundie, Jensen, Bourke & Havens, the heating contractor was Robert Gordon, Inc., and the ventilating contractor the Haines Co.

A description in pictures of the new world headquarters building of the Kraft-Phenix Cheese Corp., which houses executive and general offices, research laboratories, and a factory producing several products for the midwest



Corrosion in Steam Heating Systems

Dissolved Gases—The Basic Causes for Corrosion

By Leo F. Collins* with Everette L. Henderson†

THERE are substantial reasons for believing that pure water alone will cause little corrosion of iron. It follows, therefore, that when corrosion occurs in a heating system it is due to reactions between substances dissolved in the condensate and the metal of the confining surfaces.

Oxygen and carbon dioxide are the only corrosion producing solutes normally found in condensates, as will be shown later. It is accepted that *each must be in solution* before it becomes aggressive. Thus, the basic factors involved in heating system corrosion are oxygen, carbon dioxide, and water. Modifying factors, such as metal heterogeneity, time-rate of condensation, time of contact, temperature, stray electric currents, galvanic action, and others are also influential, but only so because they supplement the basic factors involved.

There are two possible sources of deleterious gases —entrainment with the steam or infiltration into heating systems. Most pipe failures occur in condensate lines immediately adjacent to condensing equipment—especially those operating continuously at pressures above atmospheric and at high condensation rates. At such locations infiltration of air is improbable, yet to some it has always seemed unlikely that the small amounts of gas ordinarily entrained by steams are sufficient to cause as rapid corrosion as is frequently manifested. To throw some light on this important point a special apparatus was developed to measure the corrosiveness of condensates containing only entrained gases.

Measurement of Influence of Entrained Gases Alone

The apparatus is assembled as shown in Fig. 4. The whole system, from the point where steam is obtained up to the trap, is made of brass to minimize the possibility of deactivating the condensate by contact with ferrous surfaces before the test specimens are reached. A positive pressure of about 5 lb ga is continuously maintained; this prevents infiltration of air and also keeps the gas ladened condensate at a temperature higher than in most heating systems. When all the entrained gases are dissolved, therefore, the dissolution of the test specimens is an index of the maximum potential corrosiveness of the steam under test.

Thus far four different steams have been tested. Some difficulties have been experienced in getting all the entrained gases dissolved in the hot condensates. This is unimportant because the amount of gas dissolved in the condensate, rather than that present in the steam, has been used in correlating the data shown in Fig. 5. The curves thus may be regarded as reflecting the potential corrosiveness of any steam containing the amount of gas indicated.

The steams from plants *A* and *B* contain only CO_2, that from plant *D* oxygen alone, while that from plant *C* contains both gases. This is an important relationship to be kept in mind when studying Fig. 5.

The data show quite clearly that when oxygen is the controlling factor, as for curve *A*, corrosion reaches a maximum and thereafter remains quite constant regardless of the amount of solution flowing. This seems to be generally true of oxygen attack as has been demonstrated elsewhere on several occasions. Conceivably it is due to the protectiveness of the corrosion products formed, and it suggests that oxygen attack is largely a function of gas concentration.

When CO_2 is the controlling factor, as for all the other curves, it is indicated that corrosion is a function of both the carbonic acid concentration and the amount of solution passing. It is clear that CO_2 concentrations above 2 ppm can cause serious corrosion if adequate quantities of condensate are flowing.

A comparison of the data for plant *C* with those for plants *A* and *B* indicates that small amounts of oxygen (which were present in the case of plant *C*) at least exert no measurable influence upon the reactivity between carbonic acid and iron.

A study of curves *B*, *C*, *D*, *E*, and *F* will readily show that the amount of iron dissolved is not stoichiometrically** equivalent to the amount of carbonic acid which passed over the test specimens. Nevertheless it is indicated that CO_2 is more active than has been considered true heretofore and certainly a much more impor-

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Part 2. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, September, 1939, pp. 539-542.

**Complete reaction of all the CO_2, according to the equation $Fe + 2H_2CO_3 = Fe(HCO_3)_2 + H_2$, did not take place.

This comprehensive treatment of the subject of corrosion in steam heating systems is based upon about 15 years of research work for The Detroit Edison Co. and is believed to be the most authoritative treatise on the subject that has ever been prepared. . . . Engineers concerned with operation and maintenance of building heating systems, district heating engineers, and industrial piping engineers, will find the information as useful as will designers and installers of heating plants. . . . Because of its length it is necessary to publish this treatise in serial form. Accordingly, the reader is cautioned not to pass final judgment on any statement or data until the publication of the series is completed

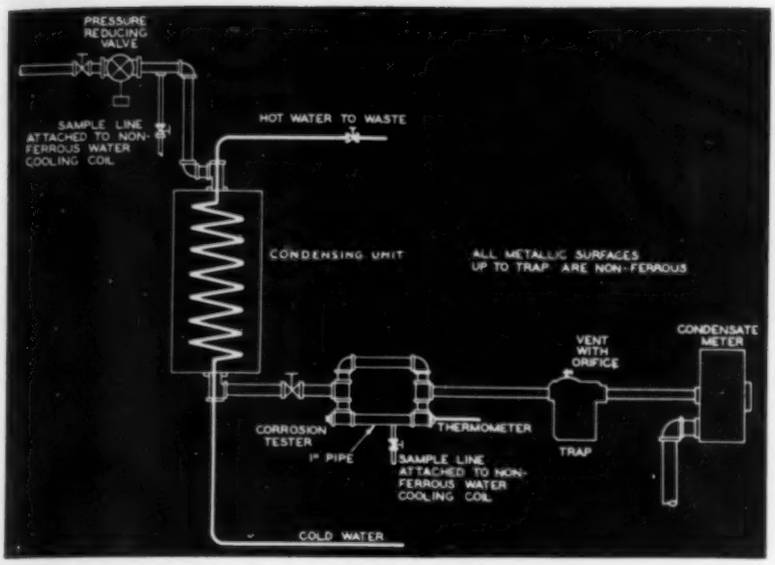


Fig. 4—Arrangement of apparatus to measure corrosion produced by entrained gases alone

tant factor than was indicated by the laboratory studies of Finnegan and his co-workers[2]. These observtions were very disconcerting until a theory was developed which now seems to offer a feasible explanation.

A New Theory of Heating System Corrosion

The theory may be briefly stated as follows: When the pH value†† of a condensate exceeds an approximate value of 5.9 the intrinsic rate of corrosion is governed principally by the amount of dissolved oxygen present. When the pH value is below 5.9 corrosion is governed principally by the H-ion concentration and "active" corrosion does not occur unless appreciable quantities of condensate are flowing. This theory was suggested by the data shown in Fig. 6.

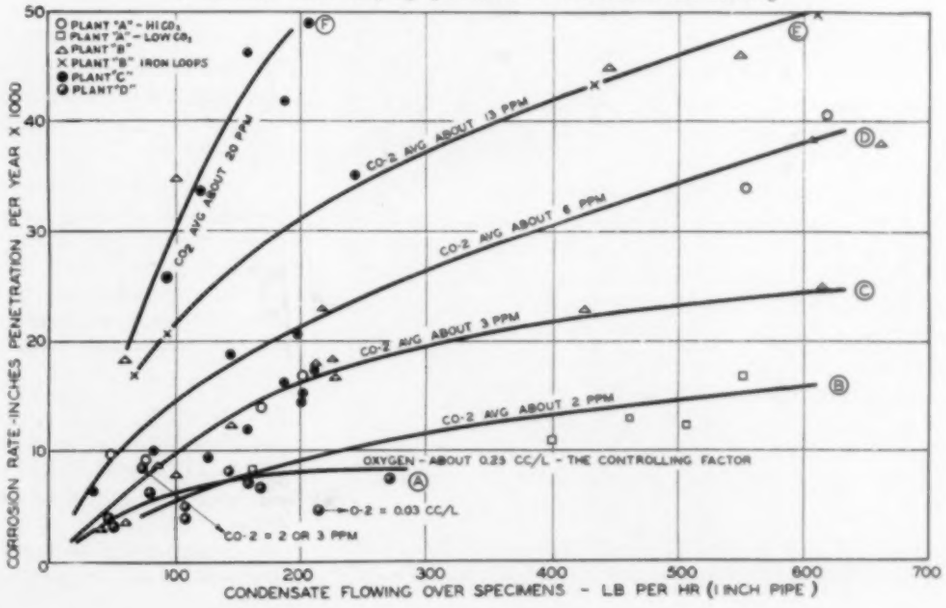
Curve *A* shows the calculated CO_2—pH relationship which exists in water containing only varying quantities of carbon dioxide. Curve *B* shows the average of a large number of CO_2 and pH determinations made on samples of steams condensed in copper cooling coils to room temperature. Curve *C* shows the average of such determinations with samples of condensate from operating heating systems. Curve *D* is the mean of Finnegan's published data for his recirculating laboratory system.

Curve *A* is theoretical and therefore not subject to experimental errors. Any deviation of experimental data from this curve can be ascribed to analytical errors or dissolution of materials which tend to raise the pH value while not materially affecting the total CO_2 concentration. When steams are condensed in copper coils and introduced into glass bottles it is well known that dissolution of small amounts of the contacting surfaces takes place. This raises the pH value and thus accounts in large measure for the deviation of curve *B* from the theoretical. With respect to curve *C* it is evident that since the condensates were in contact with ferrous surfaces some iron was dissolved. This, as will be shown later, increases the pH value and accounts for the deviation of curve *C* from the theoretical. With respect to curve *D* the original investigators stated: "During the progress of the tests the solutions became saturated with respect to ferric hydroxide." If Finnegan's solutions were saturated, it may be that curve *D* represents the limits of such action by iron compounds.

From these curves it can be reasoned that at the instant condensates are formed from CO_2 ladened steams, their CO_2-pH relationship will be as indicated by curve *A*. As they contact iron surfaces, this relationship will approach that shown by curve *D*. If the conditions of dissolution are such as to make the resulting solutions initially definitely acidic, acid corrosion will ensue.

It is a well established principle of chemistry that the hydrogen-ion concentration of any acid solution determines the strength (or the acidity) of the solution. It is also established that all acid solutions of equal H-ion concentration will dissolve the same quantity of metal provided no oxidizing action of the acid upon the metal occurs. Whitman, Russell and Altieri[3] have shown that steel will be attacked by carbonic acid, with

Fig. 5—The relation between corrosion and gas content of *condensate* when varying quantities of condensate are flowing


PLANT "A" - HI CO2
PLANT "A" - LOW CO2
PLANT "B"
PLANT "B" - IRON LOOPS
PLANT "C"
PLANT "D"
CO2 AVG ABOUT 20 PPM
CO2 AVG ABOUT 13 PPM
CO2 AVG ABOUT 6 PPM
CO2 AVG ABOUT 3 PPM
CO2 AVG ABOUT 2 PPM
OXYGEN - ABOUT 0.23 CC/L - THE CONTROLLING FACTOR
O2 = 0.03 CC/L
CO2 = 2 OR 3 PPM
CORROSION RATE-INCHES PENETRATION PER YEAR X 1000
CONDENSATE FLOWING OVER SPECIMENS - LB PER HR (1 INCH PIPE)

[2]T. J. Finnegan, R. C. Corey, and D. D. Jacobus, *Industrial and Engineering Chemistry*, Vol. 27, No. 7, July, 1935.

††The hydrogen-ion concentration is a measure of the acidic properties of a solution. The pH value, which is the negative logarithm of the hydrogen-ion concentration, is a convenient method adopted for expressing this function.

the evolution of hydrogen, beginning at pH 5.9 at room temperature and at higher pH values as the temperature is increased.

All of these observations, when considered collectively, indicate that so long as the pH value of a carbonic acid solution stays below pH 5.9 corrosion of steel will ensue at the same rate as for solutions of hydrochloric and sulphuric acids of the same H-ion concentration. Unlike these mineral acids, however, carbonic acid is weakly ionized. Its pH value begins to rise at a more rapid rate than occurs with the strong acids. It will, therefore, not cause the same amount of dissolution as will an *equivalent weight* of the mineral acids mentioned. But, if freshly formed condensate of a pH value below 5.9 continuously makes contact with a given location, acid corrosion necessarily must result.

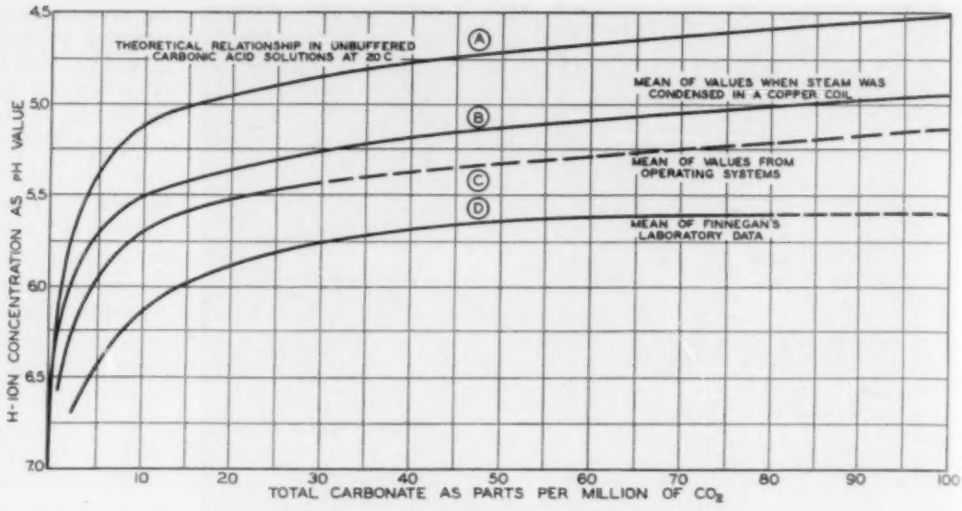


Fig. 6—Curves showing the CO_2-pH relationships of condensates "buffered" to different degrees

Curve *A*, Fig. 6, shows if more than about 2 ppm of CO_2 are dissolved in a freshly formed condensate a pH value lower than 5.9 will eventuate. Obviously then, the first metal surfaces with which such condensates make contact will experience acid attack. The initial rate of corrosion will depend upon the CO_2 concentration of the condensate and the quantity of condensate that contacts the pipe walls. Since only a small amount of dissolved iron will raise the pH value considerably, it follows that acid corrosion will occur only reasonably close to the location where condensate forms. Beyond this point, however, corrosion may occur if dissolved oxygen is also present. Thus, when a steam containing CO_2 is used it is indicated that condensing equipment and the return lines immediately following will be attacked unless constructed of a material which resists acid corrosion. In the latter case, acid corrosion will occur after the passive metal has been traversed.

[To be continued]

[a]W. G. Whitman, R. P. Russell, and V. J. Altieri, *Industrial and Engineering Chemistry*, Vol. 16, No. 7, July, 1924.

Selection of Heating Design Temperatures Is Analyzed in Carnegie Institute Bulletin

An analysis of winter temperatures for 120 cities which provides a basis for the selection of heating design temperatures has been published as Engineering Bulletin No. 82 by Carnegie Institute of Technology, Pittsburgh, Pa., and may be obtained from the Institute for 50 cents. It was prepared by Clark M. Humphreys, Assistant Professor of Mechanical Engineering, Theodore Ahrens Professorship, who was assisted by workers furnished by the Works Progress Administration.

The booklet discusses the importance of design temperature, modifying conditions, effect of wind, present methods of determining outside design temperatures, the scope of the study and its results. Tables are presented which include data on the length of the heating season for the various cities, the number of times during the period of study that various daily mean temperatures were recorded, daily minimum temperatures, frequency and recurrence of daily minimum temperatures, average number of times per year that daily minimum and daily mean temperatures have been as low as or lower than various specified temperatures, averages of monthly mean temperatures for December, January and February, etc.

Following are the conclusions of the study as given in this bulletin:

1) The daily mean temperature is a better indication of heating load than the daily minimum temperature. It is therefore a more logical basis for the determination of design temperature.

2) The design temperature should be based upon the probable frequency of recurrence of low temperatures, rather than upon the lowest daily mean or daily minimum temperature on record.

3) There is no one frequency of recurrence which should be used for the selection of all design temperatures. It will depend upon the type of building or the purpose for which it is to be used, upon the severity of the weather in the district in which the building will be built, and upon local weather peculiarities.

4) Any attempt to predict weather conditions in one city from weather data for another city should be made with caution.

How to Weld Heavy Wall Pipe

With the Single Flame, Multi-Layer Technique

By M. R. Scott*

FOR welding industrial piping in the heavier wall thicknesses, a new technique has been developed which has a number of advantages; it is known as the single flame, multi-layer technique.

Depositing the weld metal in layers has two advantages over single layer welding: (1) the average operator is less likely to find laps and inclusions in the finished welds; (2) the ductility of the first layer is increased by the normalizing or grain refining action produced as additional layers are laid down. The simplicity of the single flame and backhand techniques are further reasons why the process requires no special operator training. The backhand technique makes it easy for the operator to control the molten weld metal at all times, regardless of piping position, size, or wall thickness. In addition, each layer can be laid to such a contour that no chipping is necessary prior to laying subsequent layers.

Multi-layer welds which will more than meet code requirements can readily be made by the average operator using the single flame, backhand technique.

Specifications for welded piping have been set up by a number of organizations. These standards, or specifications, are such that both the operator and the process must qualify, and, while there are variations between the different codes, still they are essentially the same.

For the qualification of the process, it is necessary that the joint be made in the positions which will be encountered in actual construction, and welding must be done in accordance with the particular specifications. Results of numerous tests show that multi-layer welds made with the single flame, backhand technique, have physical properties equal to or better than those required by these codes.

Joint Design

Multi-layer welding is recommended for pipe wall thicknesses of ⅜ in. and over. As shown by the accompanying sketch, a 60 deg included angle, with shoulders of approximately $\frac{3}{32}$ in., is recommended for all sizes of pipe.

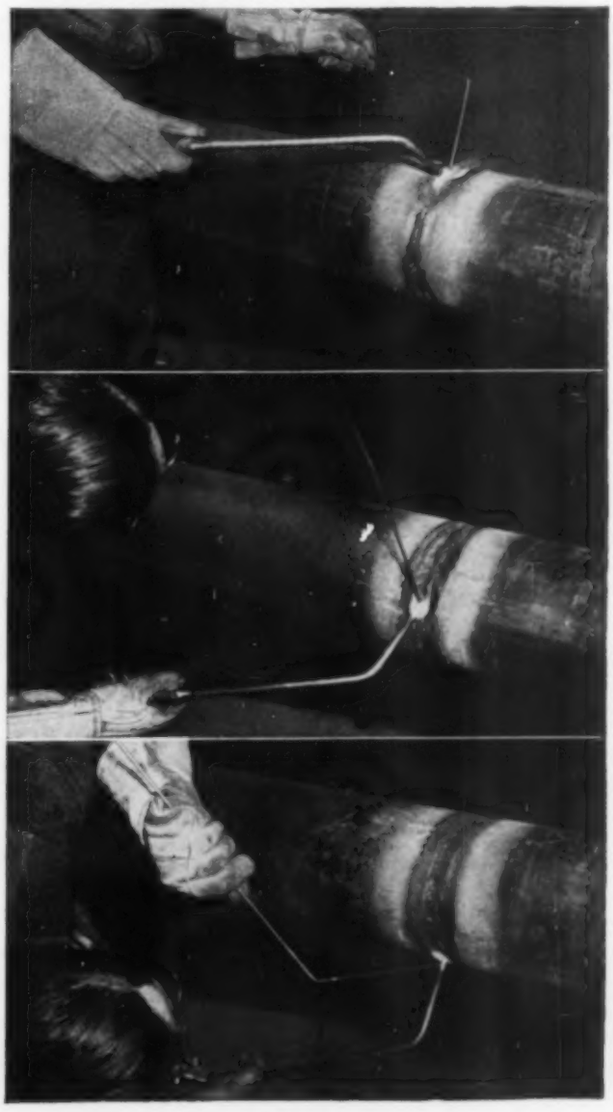
The angle of bevel is important because it has a definite effect upon the ease, quality, and cost of welding. Formerly, joints with included angles as large as 90 deg were used. Since the introduction of the backhand technique, the trend has been toward a narrower vee and consequent greater ease of welding. In fact, in the heavier wall thicknesses (⅝ in. and ¾ in.) skilled operators can obtain excellent results with a 50 deg included angle. However, average operators will find the 60 deg angle preferable for the general run of work.

The shoulder, or straight section of the joint, is also important, and should be $\frac{3}{32}$ in. plus or minus $\frac{1}{32}$ in. for most satisfactory results. If shoulders are omitted, the feather edge melts away during welding, thus, in effect, destroying the spacing. In addition, the beveled edges are apt to become damaged during shipping or moving the pipe.

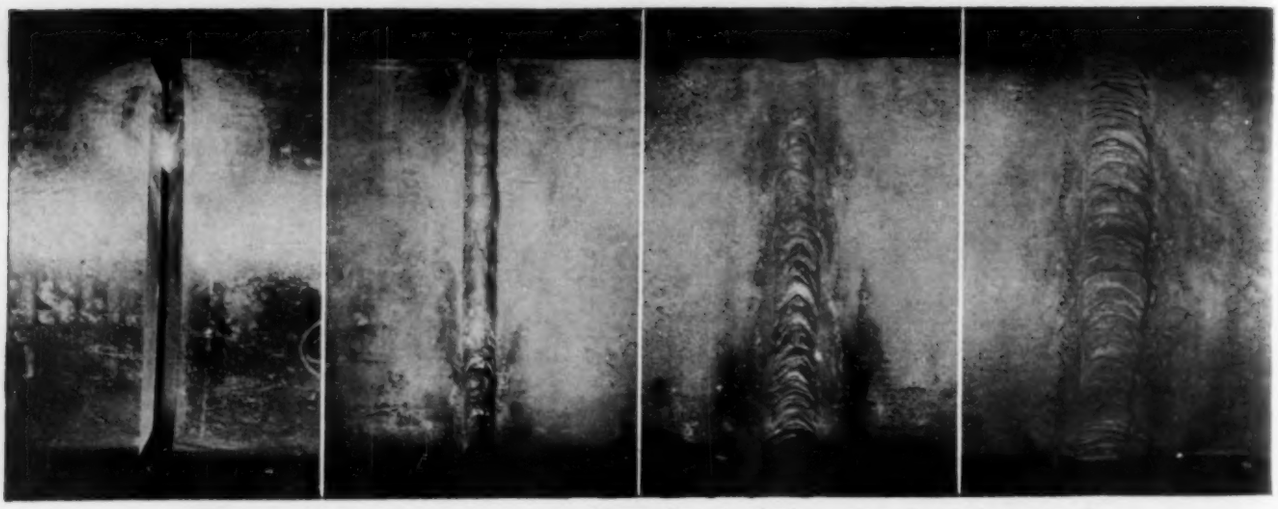
Spacing

The abutting edges of all butt joints should be separated a sufficient amount before tack welding to provide for contraction of the tack welds and to insure an

These pictures show, from top to bottom, the positions of rod and blowpipe for flat, vertical and overhead welding



*Service Engineer, The Linde Air Products Co.



Steps in the multi-layer welding of 8 in. pipe with 5/8 in. wall thickness. Left to right: tack welded, first layer deposited, second layer deposited, and completed weld showing the third layer

average spacing of 3/32 in. during the welding operation. A spacing of more than 1/8 in. during welding will cause increased rod consumption and, under certain circumstances, welding will be more difficult. If the spacing

Table 1—Spacing for Line Joints Before Tack Welding

Pipe Diameter, in.	2 to 3	4 to 6	8 to 12	14 and over
Spacing, in.	3/32	1/8	3/32	1/16

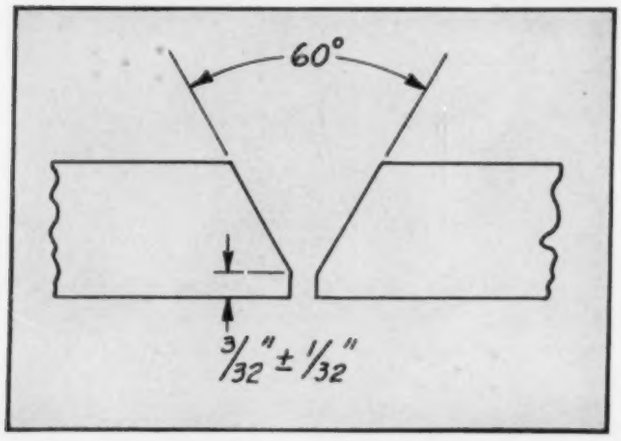
is less than 1/16 in., it will be more difficult to secure fusion to the bottom of the vee. Table 1 can be used as a guide for spacing line joints before tack welding.

Welding Rod

High test steel welding rod is recommended for the multi-layer welding of heavy wall piping. This rod is particularly well adapted to position welding since it permits a relatively large puddle to be carried in all positions without the danger of the molten metal flowing out of control.

The size of the rod to use will depend somewhat on the skill of the operator. As a general rule, it is advisable to use the largest diameter rod which can be readily handled. This prevents overheating the puddle and allows better control in using the rod as a means of controlling puddle temperature. However, with the larger rods, there is always the danger of an inexperienced operator depositing metal in the semi-plastic condition. A 1/16 in. diameter rod is a good average size for the multi-layer position welding of wall thicknesses from 3/8 in. to 3/4 in.

A 60 deg included angle should be used for multi-layer welds in heavy wall piping



Flame Adjustment

A neutral or very slight excess acetylene flame should be used for multi-layer pipe welding. Although a neutral flame is preferred, a slight excess of acetylene will be found to increase the ease of welding without a noticeable loss in the ductility of the welds.

For most work, soft, low velocity, bulbous flames in large tips are recommended for position welding. The tip size is dependent on the wall thickness and rod size and should be determined after the other variables have been fixed. In general, a No. 10 or No. 11 tip is most satisfactory for a 3/8 in. wall thickness, while a No. 13 tip should be used for a 3/4 in. wall thickness. No. 11 or 12 tips should be used for intermediate sizes.

Tack Welds

Tack welds should serve two purposes: (1) to maintain the desired distances between abutting edges; and (2) to permit longitudinal alignment of the pipe. Tack welds to serve these purposes usually require very little strength, and on this basis most of them, as commonly made, are far larger than required. They are too deep, too long, and too numerous. All that is required on 8 in. pipe, for instance, is three equally spaced tack welds 1 in. in length and, in depth, one-half the wall thickness of the pipe. The edges should be concave and well fused to the base metal.

Principles of Puddle Control

As pipe welding in general and position welding in particular involve welding in all positions—flat, vertical, overhead, and horizontal-vertical (peripheral welds on vertical pipe)—a thorough knowledge of the factors which permit control of the weld metal in these positions will aid in a better understanding of the multi-layer technique.

The most important force at work on the molten metal

is gravity, and its effect must be restrained by one or more of four means: (1) manipulating the rod in the molten puddle, (2) damming the puddle with the flame, (3) damming the puddle with cooled weld metal and base metal, and (4) regulating the temperature and size of the puddle in such a way that its cohesiveness or surface tension will maintain it under control.

Overhead Welding

Very little difficulty will be encountered by the operator in flat welding since this is by far the simplest welding position. However, a few pointers will prove helpful in producing maximum quality welds in the three other positions.

When welding directly overhead, it is possible to maintain a surprisingly large puddle of molten metal if just one precaution is observed. Everyone knows that water or oil sprayed lightly on the underside of a flat surface will fall only after a complete drop has taken shape. Likewise, if the welding puddle is kept from forming into a drop, cohesion or surface tension prevents its fall. Overhead welding is not difficult if the pool of metal is kept small and not too fluid.

Vertical Welding

When welding vertically, as when bringing the puddle down the side of a pipe joint, the influence of surface tension in maintaining the puddle decreases. This is because the same amount of liquid that could be suspended on the underside of a flat surface will collect and run on a vertical surface. Consequently, the flame dam (that is, the pressure exerted by the burning gases) must be relied upon to a considerable extent in maintaining a puddle of fair size. Vertical welding is the most difficult of all the positions since the danger of laps and lack of fusion to the sides of the vee is always present. These effects can be minimized by carrying a shallow pool of metal through this position.

Horizontal-Vertical Welding

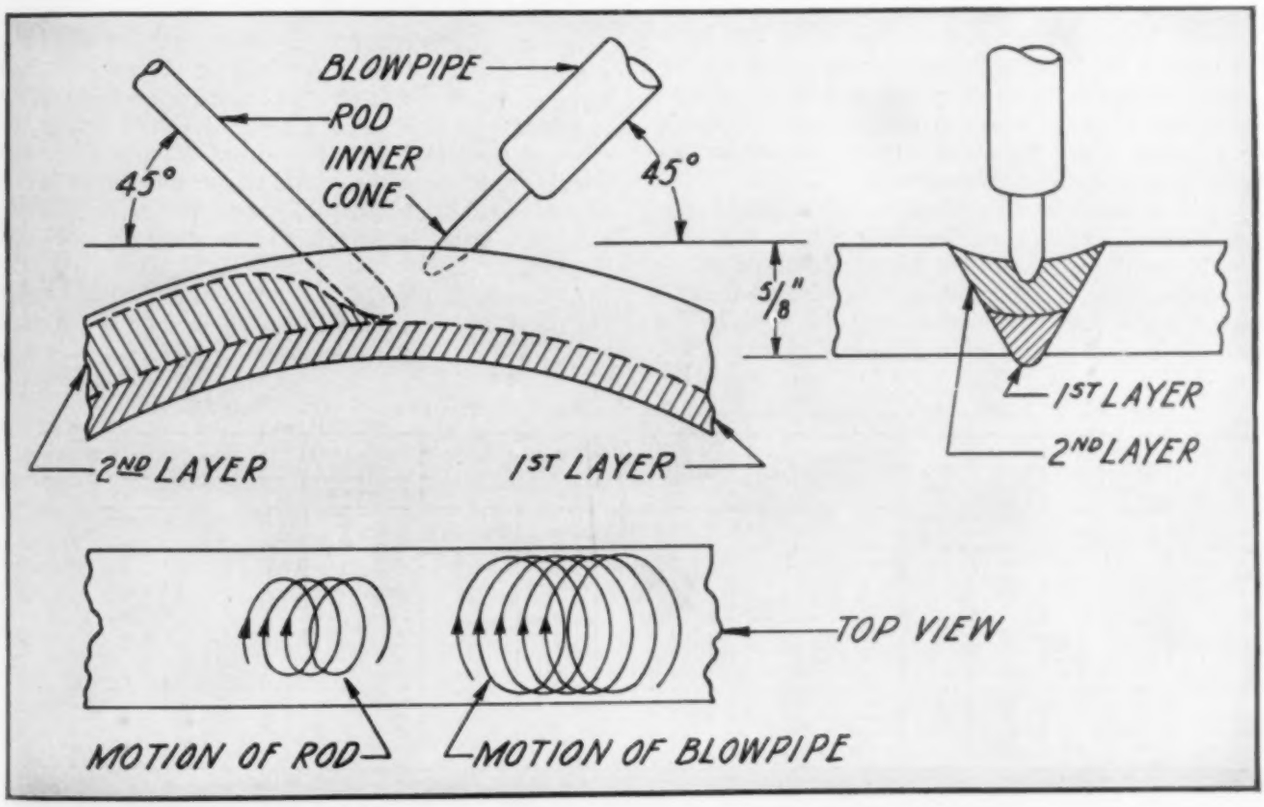
Horizontal-vertical welding (where the pipe is vertical and the joint, horizontal) is probably second to vertical welding in the amount of skill required to make a satisfactory weld.

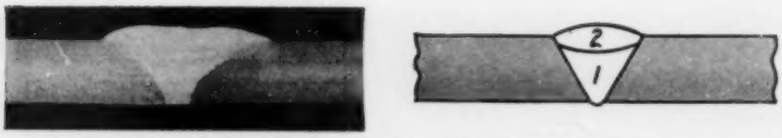
In making the weld, and still using the backhand technique, of course, the flame should be pointed ahead of the rod and should be directed mainly on the lower half of the vee. Weld metal should be built up along the lower side of the vee somewhat in advance of the upper side thereby providing a shelf or ledge for the molten puddle which is flowed in a diagonal direction across the weld. The upper edge of the vee is not melted unduly and a good rate of speed can be maintained. Should the flame be pointed directly into the weld puddle, the tendency would be for the metal to collect in a large puddle at the bottom of the vee and then fall down the side of the pipe.

Multi-Layer Technique

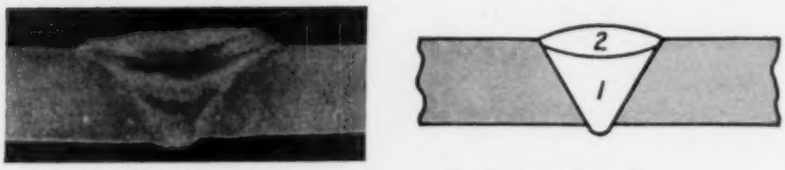
The photographs and sketches on page 626 indicate the correct layer thicknesses and contours for the multi-layer welding of various pipe wall thicknesses. Two

In multi-layer welding, both rod and flame should be moved in a circular direction with the flame following the arc of the larger radius

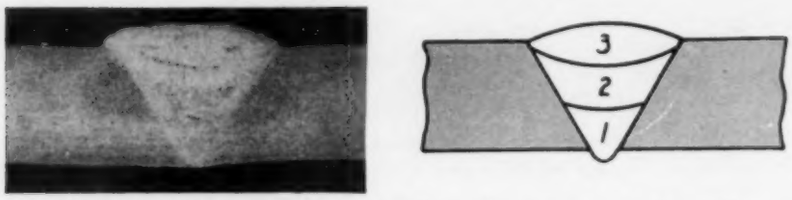




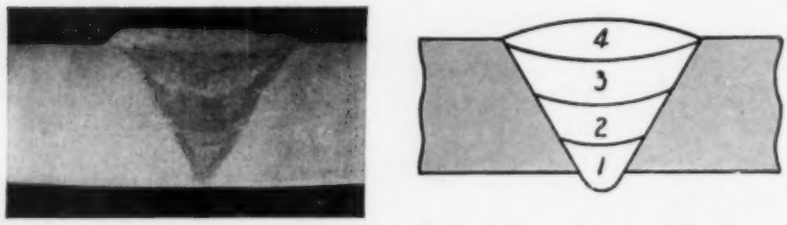
⅜ in. wall—2 layers. 1st pass, ¾t. 2nd pass completes weld



½ in. wall—2 layers. 1st pass, ¾t. 2nd pass completes weld



⅝ in. wall—3 layers. 1st pass, ¼t. 2nd pass, ¾t. 3rd pass completes weld



¾ in. wall—4 layers. 1st pass, ¾t. 2nd pass, ½t. 3rd pass, ¾t. 4th pass completes weld

These sketches indicate the correct layer thicknesses and contours for various pipe wall thicknesses. The macrographs show etched cross-sections of the welds. (In the captions, *t* represents the wall thickness of the pipe)

layers are recommended for ⅜ in. and ½ in. walls, three layers for a ⅝ in. wall, and four layers for a ¾ in. wall. Using the backhand technique the first layer of a position weld should be made by starting at the top of the joint and carrying the puddle around the pipe to the bottom, then restarting at the top and completing the opposite side. Subsequent layers should be laid similarly until the weld is completed.

It will be noted from the sketches that all base layers have concave contours, so that only a very thin skin of weld metal requires fusing when subsequent passes are made. This is important as it provides the maximum desirable heat treatment without the necessity for re-fusing a large portion of weld metal, features which are desirable from both metallurgical and economic standpoints. The concave contour will also be found advantageous from the operator's point of view because the flame spreads evenly over the surface, bringing it uniformly to the fusion temperature. On the other hand, considerable skill and effort are required to fuse the surface evenly if an undesirable convex surface meets the beveled face.

Two practices should be avoided in order to prevent the formation of a convex head. These are (1) carrying an excessively large puddle of molten weld metal which cannot be fused in properly, and (2) inadequate preheat in the base metal. These two conditions can be overcome by directing more of the welding flame on the base metal rather than the rod, and by preheating the vee a sufficient distance ahead. Both rod and flame should be moved in circular directions with the flame following the arc of the larger radius.

These conditions are more easily obtained when making the first pass because the thin, beveled edges rise in temperature much more rapidly than does the large mass of metal which must be heated up for laying subsequent layers. Hence, the first layer can be laid at relatively faster speeds than subsequent layers.

Heat Treating Weld Metal

Reheating above the critical temperature has long been a standard method for improving the impact resistance and ductility of cast steels. The same procedure applied in principle to weld metal produces identical results because of the added grain refinement acquired when the steel is raised above the critical temperature and allowed to cool slowly. Additional heating and cooling operations still further improve the physical characteristics of the metal.

Although the physical properties of multi-layer welds are equal or better than those required by the various

[*Concluded on page* 628]

Table 2—Speed and Consumption Figures for the Multi-Layer Welding of 8 in.[a] Heavy Wall Piping

Type of Weld	Pipe Size Inches		Bevel Incl. Angle, Deg	Number of Passes	Rod Size Used in Each Pass, Inches				Tip Used for Each Pass, No.				Consumption Per Weld, Cu Ft		Rod Used, Lb					Welding Time, Min				
															Per Layer				Total	Per Layer				Total
	Diam.	Wall			1	2	3	4	1	2	3	4	Oxygen	Acetylene	1	2	3	4		1	2	3	4	
Rolling	8	3/8	60	2	3/16	3/16	....	....	10	10	....	....	19	19	0.47	0.50	....	....	0.97	9	15	....	....	24
	8	1/2	60	2	3/16	1/4	....	....	12	11	....	....	33	33	0.71	0.73	....	....	1.44	14	16	....	....	30
	8	5/8	60	3	3/16	3/16	1/4	....	12	12	12	....	67	67	0.47	1.10	0.81	....	2.38	15	17	21	....	53
	8	3/4	60	4	3/16	3/16	1/4	1/4	13	13	13	13	119	119	0.56	0.86	0.89	1.20	3.51	16	24	20	22	82
Position	8	3/8	60	2	3/16	3/16	....	....	11	10	....	....	31	31	0.47	0.75	....	....	1.22	14	20	....	....	34
	8	1/2	60	2	3/16	3/16	....	....	12	11	....	....	54	54	0.56	0.97	....	....	1.53	20	30	....	....	50
	8	5/8	60	3	3/16	3/16	3/16	....	12	12	11	....	86	86	0.52	1.12	0.62	....	2.26	22	24	30	....	76
	8	3/4	60	4	3/16	3/16	3/16	3/16	13	13	13	12	164	164	0.70	0.75	1.32	0.85	3.62	26	28	30	40	124
Horizontal-Vertical	8	3/8	60	2	3/16	3/16	....	....	11	10	....	....	22	22	0.48	0.55	....	....	1.03	9	17	....	....	26
	8	1/2	60	2	3/16	3/16	....	....	12	11	....	....	34	34	0.45	0.86	....	....	1.31	13	18	....	....	31
	8	5/8	60	3	3/16	3/16	3/16	....	12	12	11	....	87	87	0.56	0.71	1.22	....	2.49	18	26	34	....	78
	8	3/4	60	4	3/16	3/16	1/4	1/4	13	13	13	12	168	168	0.49	0.52	1.10	1.35	3.46	18	22	28	52	120

[a]Data for pipe sizes other than 8 in. can be obtained by multiplying rod and gas consumptions and speeds by the ratio of the diameters.

Properties of Mixtures of Air and Saturated Water Vapor for Barometric Pressures from 22 In. to 32 In. of Mercury

By William Goodman*

Barometric Pressure 25 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s'		h_v	h
1	2	3	4	5	6	7	8	9
— 40	0.0001393	0.9750	12.66	0.02239	0.2401	0.1478	0.1453	—9.455
— 39	.0001474	1.032	12.69	.02370	.2401	.1564	.1539	—9.206
— 38	.0001562	1.094	12.72	.02511	.2401	.1657	.1631	—8.957
— 37	.0001654	1.158	12.75	.02659	.2401	.1755	.1727	—8.707
— 36	.0001751	1.226	12.78	.02815	.2401	.1858	.1829	—8.457
— 35	0.0001851	1.296	12.81	0.02976	0.2401	0.1964	0.1935	—8.207
— 34	.0001959	1.371	12.84	.03150	.2401	.2079	.2049	—7.955
— 33	.0002071	1.450	12.87	.03330	.2401	.2197	.2167	—7.703
— 32	.0002190	1.533	12.90	.03521	.2401	.2324	.2292	—7.451
— 31	.0002313	1.619	12.93	.03718	.2401	.2454	.2422	—7.198
— 30	0.0002445	1.712	12.96	0.03931	0.2401	0.2594	0.2561	—6.944
— 29	.0002582	1.808	12.99	.04152	.2401	.2740	.2706	—6.689
— 28	.0002725	1.907	13.02	.04380	.2401	.2891	.2857	—6.434
— 27	.0002877	2.014	13.05	.04626	.2401	.3053	.3018	—6.178
— 26	.0003035	2.125	13.08	.04880	.2401	.3221	.3185	—5.922
— 25	0.0003201	2.241	13.11	0.05147	0.2401	0.3397	0.3361	—5.664
— 24	.0003377	2.364	13.14	.05429	.2402	.3583	.3547	—5.405
— 23	.0003562	2.494	13.17	.05727	.2402	.3779	.3743	—5.146
— 22	.0003753	2.627	13.20	.06034	.2402	.3982	.3945	—4.886
— 21	.0003955	2.768	13.23	.06358	.2402	.4196	.4159	—4.624
— 20	0.0004167	2.917	13.26	0.06700	0.2402	0.4421	0.4384	—4.362
— 19	.0004390	3.073	13.29	.07058	.2402	.4658	.4621	—4.098
— 18	.0004621	3.234	13.32	.07429	.2402	.4902	.4865	—3.834
— 17	.0004863	3.404	13.36	.07818	.2402	.5160	.5123	—3.568
— 16	.0005121	3.584	13.39	.08232	.2402	.5433	.5396	—3.300
— 15	0.0005384	3.769	13.42	0.08656	0.2402	0.5713	0.5677	—3.032
— 14	.0005664	3.965	13.45	.09105	.2403	.6009	.5973	—2.763
— 13	.0005958	4.170	13.48	.09578	.2403	.6321	.6287	—2.491
— 12	.0006262	4.384	13.51	.1007	.2403	.6644	.6611	—2.219
— 11	.0006578	4.605	13.54	.1058	.2403	.6979	.6947	—1.945
— 10	0.0006917	4.842	13.57	0.1112	0.2403	0.7339	0.7308	—1.669
— 9	.0007263	5.084	13.60	.1168	.2403	.7706	.7677	—1.392
— 8	.0007633	5.343	13.63	.1227	.2403	.8099	.8071	—1.113
— 7	.0008009	5.606	13.66	.1288	.2404	.8437	.8472	—0.8328
— 6	.0008405	5.883	13.69	.1351	.2404	.8918	.8895	—0.5505
— 5	0.0008827	6.179	13.72	0.1419	0.2404	0.9365	0.9346	—0.2655
— 4	.0009259	6.481	13.75	.1489	.2404	0.9823	0.9807	+0.0207
— 3	.0009711	6.798	13.78	.1561	.2404	1.030	1.029	0.3090
— 2	.001019	7.132	13.81	.1638	.2405	1.081	1.080	0.6000
— 1	.001069	7.480	13.84	.1718	.2405	1.134	1.133	0.8934
0	0.001120	7.837	13.87	0.1800	0.2405	1.188	1.188	1.188
1	.001173	8.210	13.90	.1886	.2405	1.244	1.245	1.485
2	.001229	8.602	13.93	.1976	.2406	1.304	1.305	1.785
3	.001287	9.011	13.96	.2070	.2406	1.366	1.368	2.088
4	.001349	9.442	13.99	.2169	.2406	1.431	1.434	2.394
5	0.001412	9.882	14.02	0.2270	0.2406	1.498	1.501	2.701
6	.001478	10.34	14.05	.2376	.2407	1.568	1.572	3.012
7	.001546	10.82	14.08	.2486	.2407	1.641	1.646	3.326
8	.001618	11.33	14.11	.2601	.2407	1.717	1.723	3.643
9	.001693	11.85	14.14	.2723	.2408	1.797	1.804	3.964
10	0.001770	12.39	14.17	0.2846	0.2408	1.878	1.886	4.286
11	.001853	12.97	14.20	.2979	.2408	1.966	1.975	4.615
12	.001937	13.56	14.23	.3114	.2409	2.055	2.066	4.946
13	.002023	14.16	14.26	.3252	.2409	2.146	2.158	5.278
14	.002116	14.81	14.29	.3402	.2410	2.245	2.258	5.618
15	0.002211	15.48	14.32	0.3555	0.2410	2.346	2.361	5.961
16	.002309	16.16	14.35	.3712	.2410	2.450	2.466	6.306
17	.002411	16.88	14.38	.3876	.2411	2.558	2.577	6.657
18	.002518	17.62	14.41	.4047	.2411	2.671	2.692	7.012
19	.002629	18.40	14.44	.4227	.2412	2.789	2.812	7.372
20	0.002743	19.20	14.47	0.4410	0.2412	2.910	2.935	7.735
21	.002864	20.05	14.50	.4605	.2413	3.039	3.066	8.106
22	.002987	20.91	14.53	.4803	.2413	3.169	3.199	8.479
23	.003117	21.82	14.56	.5011	.2414	3.307	3.339	8.859
24	.003251	22.76	14.59	.5227	.2415	3.449	3.484	9.244
25	0.003388	23.72	14.62	0.5447	0.2415	3.595	3.633	9.633
26	.003535	24.74	14.65	.5682	.2416	3.750	3.792	10.03
27	.003683	25.78	14.68	.5921	.2417	3.908	3.953	10.43
28	.003838	26.87	14.71	.6170	.2417	4.072	4.120	10.84
29	.003999	28.00	14.74	.6430	.2418	4.243	4.296	11.26
30	0.004167	29.17	14.77	0.6700	0.2419	4.422	4.478	11.68
31	.004343	30.40	14.80	.6982	.2420	4.608	4.669	12.11
32	.004519	31.64	14.83	.7264	.2420	4.795	4.862	12.54
33	.004710	32.97	14.86	.7570	.2421	4.997	5.069	12.99
34	.004905	34.33	14.89	.7883	.2422	5.204	5.281	13.44
35	0.005105	35.73	14.92	0.8206	0.2423	5.416	5.499	13.90
36	.005316	37.21	14.95	.8544	.2424	5.640	5.728	14.37
37	.005533	38.73	14.98	.8890	.2425	5.871	5.965	14.85
38	.005757	40.30	15.01	.9253	.2426	6.109	6.209	15.33
39	.005990	41.93	15.04	.9625	.2427	6.355	6.462	15.82

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s'		h_v	h
1	2	3	4	5	6	7	8	9
40	0.006230	43.61	15.07	1.001	0.2428	6.610	6.724	16.32
41	.006479	45.35	15.10	1.041	.2429	6.874	6.995	16.84
42	.006737	47.16	15.13	1.083	.2430	7.148	7.278	17.36
43	.007003	49.02	15.17	1.125	.2432	7.430	7.567	17.89
44	.007279	50.95	15.20	1.170	.2433	7.723	7.868	18.43
45	0.007569	52.98	15.23	1.216	0.2434	8.031	8.186	18.99
46	.007865	55.06	15.26	1.264	.2435	8.345	8.509	19.55
47	.008171	57.20	15.29	1.313	.2437	8.669	8.844	20.12
48	.008490	59.43	15.32	1.364	.2438	9.008	9.193	20.71
49	.008818	61.73	15.35	1.417	.2440	9.356	9.552	21.31
50	0.009161	64.12	15.38	1.472	0.2441	9.719	9.927	21.93
51	.009516	66.61	15.41	1.529	.2443	10.10	10.32	22.56
52	.009875	69.12	15.44	1.587	.2444	10.48	10.71	23.19
53	.01026	71.80	15.47	1.648	.2446	10.88	11.13	23.85
54	.01064	74.49	15.50	1.710	.2448	11.29	11.55	24.51
55	0.01105	77.32	15.53	1.775	0.2450	11.72	11.99	25.19
56	.01146	80.24	15.56	1.841	.2452	12.16	12.45	25.89
57	.01189	83.23	15.59	1.910	.2454	12.62	12.92	26.60
58	.01234	86.37	15.62	1.982	.2456	13.09	13.41	27.33
59	.01279	89.56	15.65	2.055	.2458	13.57	13.92	28.08
60	0.01327	92.86	15.68	2.132	0.2460	14.07	14.43	28.83
61	.01376	96.30	15.71	2.211	.2462	14.60	14.92	29.61
62	.01427	99.90	15.74	2.292	.2464	15.14	15.54	30.42
63	.01480	103.6	15.77	2.376	.2467	15.70	16.12	31.24
64	.01533	107.3	15.80	2.463	.2469	16.26	16.70	32.06
65	0.01590	111.3	15.83	2.552	0.2472	16.87	17.33	32.93
66	.01647	115.3	15.86	2.645	.2474	17.47	17.96	33.80
67	.01707	119.5	15.89	2.741	.2477	18.11	18.62	34.70
68	.01768	123.8	15.92	2.840	.2480	18.76	19.30	35.62
69	.01831	128.2	15.95	2.941	.2482	19.43	20.00	36.56
70	0.01897	132.8	15.98	3.047	0.2485	20.13	20.72	37.52
71	.01965	137.5	16.01	3.155	.2488	20.85	21.47	38.51
72	.02035	142.5	16.04	3.268	.2492	21.59	22.25	39.53
73	.02107	147.5	16.07	3.384	.2495	22.36	23.05	40.57
74	.02182	152.7	16.10	3.504	.2498	23.15	23.87	41.63
75	0.02258	158.1	16.13	3.626	0.2502	23.96	24.72	42.72
76	.02338	163.6	16.16	3.754	.2505	24.80	25.60	43.84
77	.02421	169.5	16.19	3.887	.2509	25.69	26.52	45.00
78	.02505	175.4	16.22	4.023	.2513	26.58	27.45	46.17
79	.02592	181.5	16.25	4.162	.2517	27.51	28.42	47.38
80	0.02682	187.8	16.28	4.306	0.2521	28.46	29.41	48.61
81	.02776	194.3	16.31	4.456	.2525	29.45	30.45	49.89
82	.02871	201.0	16.34	4.609	.2529	30.46	31.51	51.19
83	.02971	208.0	16.37	4.769	.2534	31.52	32.62	52.54
84	.03072	215.0	16.40	4.931	.2538	32.59	33.74	53.90
85	0.03177	222.4	16.43	5.100	0.2543	33.71	34.91	55.31
86	.03286	230.0	16.46	5.275	.2548	34.87	36.12	56.76
87	.03398	237.9	16.49	5.454	.2553	36.06	37.37	58.25
88	.03514	246.0	16.52	5.639	.2558	37.28	38.65	59.77
89	.03634	254.4	16.55	5.832	.2564	38.56	39.99	61.35
90	0.03757	263.0	16.58	6.028	0.2569	39.86	41.36	62.96
91	.03884	271.9	16.61	6.233	.2575	41.21	42.78	64.62
92	.04014	281.0	16.64	6.441	.2581	42.59	44.23	66.31
93	.04150	290.5	16.67	6.659	.2587	44.03	45.74	68.06
94	.04290	300.3	16.70	6.882	.2593	45.52	47.30	69.86
95	0.04433	310.3	16.73	7.112	0.2600	47.04	48.90	71.70
96	.04581	320.7	16.76	7.350	.2606	48.60	50.55	73.59
97	.04733	331.3	16.79	7.593	.2613	50.22	52.25	75.53
98	.04892	342.5	16.82	7.848	.2620	51.91	54.03	77.55
99	.05055	353.9	16.85	8.110	.2628	53.64	55.85	79.61
100	0.05222	365.5	16.88	8.377	0.2635	55.40	57.71	81.71
101	.05395	377.7	16.91	8.655	.2643	57.25	59.65	83.89
102	.05575	390.2	16.95	8.941	.2651	59.15	61.66	86.14
103	.05760	403.2	16.98	9.235	.2659	61.11	63.73	88.45
104	.05950	416.5	17.01	9.542	.2668	63.13	65.86	90.82
105	0.06145	430.2	17.04	9.855	0.2677	65.20	68.04	93.24
106	.06348	444.3	17.07	10.18	.2686	67.35	70.32	95.76
107	.06558	459.1	17.10	10.52	.2695	69.59	72.68	98.36
108	.06773	474.1	17.13	10.86	.2705	71.87	75.09	101.01
109	.06996	489.8	17.16	11.22	.2715	74.23	77.59	103.75
110	0.07227	505.9	17.19	11.59	0.2725	76.68	80.18	106.58
111	.07462	522.4	17.22	11.96	.2736	79.17	82.82	109.46
112	.07708	539.6	17.25	12.36	.2747	81.78	85.58	112.46
113	.07962	557.3	17.28	12.76	.2758	84.47	88.43	115.55
114	.08223	565.6	17.31	13.18	.2770	87.24	91.36	118.72
115	0.08492	594.4	17.34	13.61	0.2782	90.10	94.39	121.99
116	.08772	614.1	17.37	14.05	.2795	93.07	97.55	125.39
117	.09060	634.2	17.40	14.52	.2808	96.13	100.79	128.87
118	.09356	654.9	17.43	14.99	.2821	99.26	104.11	132.43
119	.09660	676.2	17.46	15.48	.2835	102.49	107.53	136.09

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(Tabulations for 22 in., 23 in. and 24 in. were published in July, August and September.)

Properties of Mixtures of Air and Saturated Water Vapor—(Continued)
Barometric Pressure 25 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)		Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s'		h_v	h	t	w	w	v_D		s'		h_v	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
120	0.09979	698.5	17.49	15.99	0.2849	105.88	111.14	139.94	160	0.3939	2757.4	18.69	62.88	0.4173	417.95	445.21	483.61
121	.1031	721.5	17.52	16.51	.2864	109.36	114.83	143.87	161	.4096	2867.4	18.72	65.38	.4243	434.62	463.13	501.77
122	.1064	744.9	17.55	17.05	.2879	112.90	118.59	147.87	162	.4263	2983.8	18.76	68.02	.4318	452.26	482.10	520.98
123	.1099	769.5	17.58	17.61	.2895	116.64	122.56	152.08	163	.4437	3105.7	18.79	70.79	.4397	470.73	501.97	541.09
124	.1135	794.8	17.61	18.19	.2911	120.47	126.63	156.39	164	.4621	3235.0	18.82	73.74	.4480	490.33	523.05	562.41
125	0.1173	821.0	17.64	18.78	0.2928	124.44	130.87	160.87	165	0.4816	3370.9	18.85	76.82	0.4567	510.92	545.21	584.81
126	.1212	848.1	17.67	19.40	.2945	128.55	135.24	165.48	166	.5022	3515.7	18.88	80.12	.4660	532.88	568.84	608.68
127	.1252	876.2	17.70	20.04	.2963	132.81	139.77	170.25	167	.5240	3667.7	18.91	83.57	.4758	555.92	593.65	633.73
128	.1293	905.2	17.73	20.71	.2982	137.20	144.44	175.16	168	.5472	3830.5	18.94	87.26	.4863	580.60	620.22	660.54
129	.1336	935.2	17.76	21.40	.3001	141.75	149.29	180.25	169	.5717	4001.6	18.97	91.14	.4972	606.52	648.14	688.70
130	0.1381	966.8	17.79	22.11	0.3022	146.55	154.40	185.60	170	0.5978	4184.6	19.00	95.31	0.5090	634.27	678.03	718.83
131	.1427	999.0	17.82	22.84	.3042	151.42	159.59	191.03	171	.6255	4378.6	19.03	99.70	.5215	663.67	709.70	750.74
132	.1474	1031.5	17.85	23.59	.3063	156.34	164.84	196.52	172	.6553	4586.8	19.06	104.4	.5349	695.23	743.72	785.00
133	.1523	1066.2	17.88	24.38	.3085	161.61	170.46	202.38	173	.6872	4810.6	19.09	109.5	.5493	729.15	780.28	821.80
134	.1575	1102.2	17.91	25.20	.3109	167.05	176.27	208.43	174	.7213	5049.1	19.12	114.9	.5646	765.30	818.25	860.01
135	0.1627	1139.0	17.94	26.05	0.3132	172.64	182.22	214.62	175	0.7574	5302.1	19.15	119.7	0.5809	803.64	860.60	902.60
136	.1682	1177.3	17.97	26.91	.3157	178.44	188.41	221.05	176	.7967	5576.8	19.18	126.9	.5985	845.28	905.51	947.75
137	.1739	1217.3	18.00	27.82	.3183	184.51	194.91	227.79	177	.8391	5873.8	19.21	133.7	.6176	890.30	954.07	996.55
138	.1798	1258.5	18.03	28.78	.3209	190.76	201.58	234.70	178	.8847	6193.2	19.24	140.9	.6381	938.71	1006.3	1049.0
139	.1860	1301.7	18.06	29.74	.3237	197.30	208.57	241.93	179	.9345	6541.5	19.27	148.8	.6605	991.50	1063.2	1106.1
140	0.1923	1346.2	18.09	30.76	0.3265	204.04	215.77	249.37	180	0.9893	6924.8	19.30	157.5	0.6852	1049.6	1125.9	1169.1
141	.1989	1392.0	18.12	31.81	.3295	210.98	223.19	257.03	181	1.049	7339.5	19.33	166.9	.7118	1112.5	1193.7	1237.2
142	.2058	1440.9	18.15	32.91	.3326	218.40	231.12	265.20	182	1.114	7795.9	19.36	177.2	.7412	1181.6	1268.4	1312.1
143	.2129	1490.0	18.18	34.05	.3358	225.84	239.08	273.40	183	1.185	8295.7	19.39	188.6	.7733	1257.4	1350.2	1394.1
144	.2202	1541.7	18.21	35.23	.3391	233.67	247.46	282.02	184	1.265	8852.2	19.42	201.2	.8091	1341.7	1441.3	1485.4
145	0.2280	1595.7	18.24	36.46	0.3426	241.87	256.25	291.05	185	1.353	9468.9	19.45	215.2	0.8487	1435.2	1542.2	1586.6
146	.2361	1652.7	18.27	37.74	.3463	250.50	265.49	300.53	186	1.451	10160	19.48	230.8	0.8931	1539.9	1655.3	1700.0
147	.2445	1711.2	18.30	39.07	.3500	259.37	274.99	310.27	187	1.562	10935	19.51	248.4	0.9430	1657.5	1782.3	1827.2
148	.2532	1772.3	18.33	40.46	.3539	268.62	284.90	320.42	188	1.688	11817	19.54	268.4	0.9997	1791.2	1926.7	1971.9
149	.2622	1835.4	18.36	41.91	.3580	278.19	295.16	330.92	189	1.832	12827	19.57	291.3	1.065	1944.2	2092.1	2137.4
150	0.2718	1902.4	18.39	43.43	0.3623	288.35	306.04	342.04	190	1.999	13,990	19.60	317.7	1.139	2120.4	2282.3	2327.9
151	.2817	1971.6	18.42	45.01	.3668	298.84	317.29	353.53	191	2.192	15,347	19.63	348.4	1.227	2326.1	2504.6	2550.4
152	.2920	2043.7	18.45	46.64	.3714	309.76	329.00	365.48	192	2.421	16,950	19.66	384.8	1.330	2569.1	2767.2	2813.5
153	.3028	2119.5	18.48	48.37	.3763	321.25	341.32	378.04	193	2.697	18,878	19.69	428.4	1.454	2861.4	3083.1	3129.4
154	.3140	2198.3	18.51	50.16	.3813	333.20	354.14	391.10	194	3.033	21,230	19.72	481.8	1.605	3217.8	3468.3	3514.9
155	0.3259	2281.4	18.54	52.05	0.3867	345.80	367.67	404.87	195	3.452	24,166	19.75	548.4	1.794	3662.9	3949.4	3996.2
156	.3383	2367.8	18.57	54.02	.3922	358.88	381.75	419.19	196	3.991	27,934	19.78	633.8	2.036	4234.0	4566.8	4613.9
157	.3510	2456.7	18.60	56.06	.3979	372.37	396.23	433.91	197	4.708	32,957	19.81	747.5	2.359	4995.3	5389.4	5436.6
158	.3647	2553.0	18.63	58.23	.4041	386.96	411.90	449.82	198	5.708	39,955	19.84	906.0	2.809	6056.0	6536.0	6583.5
159	.3789	2652.4	18.66	60.49	.4105	402.03	428.10	466.26	199	6.809	47,597	19.87	1079.2	3.300	7214.3	7788.8	7836.6
									200	9.648	67,535	19.90	1530.8	4.582	10,236	11,055	11,103

How to Weld—

[*Concluded from page 626*]

codes, the ductility of the welds can be improved still further by (1) blowpipe normalizing, (2) stress relieving, or (3) a combination of both, the relative effectiveness increasing in the order listed. For blowpipe normalizing, welds should be reheated, using a large tip, to approximately 1750 F, or just above the critical. This is in the salmon or orange color range, as viewed in the dark, or just above a bright red temperature. Stress relieving should be done in a furnace at a temperature of approximately 1200 F, or dark cherry red as viewed in the dark. The metal should be held at this temperature for 1 hr for each inch of wall thickness, then allowed to cool in air to room temperature.

The applicability of these treatments to welded piping installations should be considered wherever the ultimate in the physical properties of welded joints is desired. As an example of the benefits derived, the blowpipe normalizing treatment followed by the stress relieving treatment will increase the ductility of ¾ in. weld test specimens sufficiently to allow 180 deg free bends.

In Table 2 are tabulated welding speeds and rod and gas consumption figures for multi-layer welds in 8 in. diameter pipe. The figures presented are for position welds, horizontal-vertical welds, and rolling welds in wall thicknesses ranging from ⅜ in. to ¾ in. The table is set up in such a manner that data for pipe sizes other than 8 in. can readily be obtained by multiplying speeds, rod or gas consumption by the ratio of pipe diameters.

It should be emphasized that the results as set forth in this table were obtained by average operators who had average ability in all types of welding, but had not specialized in any one method such as heavy wall pipe welding. In other words, the figures are representative of what can be expected in the field with operators untrained in the multi-layer welding technique. Skilled operators can obtain faster speeds than those shown in the table.

At the new nursery school being erected on the estate of Mrs. Dwight W. Morrow, Englewood, N. J., the floors are to be heated by unit heaters suspended under them so that the children may play on the floor without danger of catching cold. Each room is to have individual air conditioners built into recesses in the walls to avoid possibility of contaminated air being circulated from room to room.

The system is to be completely automatic, with a time clock arrangement to turn it on and off.

Heating-Piping-Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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BULLETIN BOARD

Welcome to Fall Meeting in Atlanta — October 30-31

CHAPTERS

The fall program of Local Chapter meetings starts actively in October, and 28 Chapters are issuing announcements about their meeting plans for the 1939-40 season.

• • •

ATTENTION

Membership address forms are being sent to all members with a request that necessary corrections be furnished by October 31, so that the Roll of Membership to be published in THE GUIDE 1940 will be up-to-date.

• • •

REMINDER

It is wise to pay dues within the calendar year to avoid becoming in arrears because *Section 6*, Article B-IV, of the By-Laws, says, "All dues shall be payable in January of each year in advance . . . .", and *Section 2*, Article B-II, says, "Any member whose dues shall remain unpaid for one (1) year shall at the discretion of the Council forfeit the privileges of membership . . . .". Already dues have been paid by 79 per cent of the membership. Payment in full or in installments convenient to the members is equally acceptable.

Come to Cleveland for the 46th Annual Meeting, January 22-26, 1940, and the 6th International Heating and Ventilating Exposition

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A Standard Air Filter Test Dust

By Frank B. Rowley* and Richard C. Jordan** (*MEMBERS*), Minneapolis, Minn.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Graduate School of the University of Minnesota in cooperation with the Engineering Experiment Station, University of Minnesota.

THIS paper presents the results of research which supplement the work described previously in papers[1, 2] delivered before the Society. The object of this part of the program was to study the requirements of a standard test dust for filters to be tested on a weight basis, and to investigate the properties of different dusts in relation to those requirements. The test apparatus and procedure were described in the previous papers. The apparatus, with the exception of the dust feeding device, was the same as that described in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. The photograph, Fig. 1, shows a general view of the test equipment as set up and used throughout the tests.

The four filters selected in the previous test program as typical of those in practice were used throughout these tests. These filters have been designated by the letters *A, B-2, C,* and *D* and were described in detail in the first paper. Briefly they may be described as follows:

A—A permanent type of cleanable oil filter, 4 in. thick, with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

B-2—A viscous coated throw-away type of filter, 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side. This filter is designated in this report as *B-2* because slight changes had been made in the oiling and fiber diameter from those filters used in the previous reports. These changes, however, did not appear to make any appreciable difference in the test results.

C—A cellular type filter, 2 in. thick. The filter was built in two sections, each with the axis of cells set at 45 deg to the center line of duct and at 90 deg to each other. The cells on the entering side were of larger dimension than those on the leaving side of the filter.

D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordian plaited in frame to give an area of approximately 12 times the cross-sectional area of air stream.

There are two fundamental requirements of any dust that is to be used for filter test work. First, it must simulate air-borne dust in all of its properties which may affect filter performance. Second, it must be possible and practical to obtain the dust in sufficient quantities for test purposes and with reasonable certainty that all of the properties can be duplicated.

The properties of a test dust which must be comparable to those of air-borne dust may be summed up as follows:

1. It must be similar in density and air flotation characteristics.
2. It must have reasonable likeness in range and distribution of dust particle size.
3. It must not have a tendency to agglomerate and thus change the physical characteristics of the dust.
4. It must not create any physiological or other hazard.

The standard filter test dust selected for the ASHVE Air Filter Code consists of 50 per cent by weight of powdered lampblack containing a minimum of 97.5 per cent free carbon and having a minimum bulk density of 3.5 lb per cubic foot, mixed with 50 per cent by weight of Pocahontas bituminous coal ash screened to pass a 200-mesh screen. There are many kinds of dust in the air with wide variations in density, particle size and shape, as well as other characteristics. Previous investigations carried out in this test program have shown that many of these factors have a marked effect on filter performance, and that a given dust may not have a like

Fig. 1—Assembled view of test apparatus

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**Instructor, Engineering Experiment Station, University of Minnesota.

[1] Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

[2] Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by Frank B. Rowley and Richard C. Jordan. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1939.

For presentation at the Fall Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlanta, Ga., October, 1939.

Table 1—Materials Investigated for Artificial Test Dusts

Type of Dust	Reasons for Discarding from Consideration
Fuller's Earth	Absorptive when used in large percentages
Air Floated Silica (325-mesh)	High density—physiological hazard.
Borax	Tendency to agglomerate and fall to bottom of test duct
Rosin Dust	Too sticky
Chalk Dust	High density
Alum	High density
Ground English Walnut Shells (parting dust)	Apparently too absorptive—Filter arrestance dropped off rapidly in test
TiO_2	Plugged feeder—Tendency to agglomerate and fall to bottom of test duct
Talc and soapstone parting dust	High density
Lycopodium	Average particle size too large
Al_2O_3	High density
Pumice	High density
Wood Ash	Too difficult to obtain in quantities

Table 2—Source of Sample and Density of Test Ashes
(Coal burned at uniform rate)

District	Coal Sample	Dust Density in Grams/cc (Jolted)	Source
West Virginia	A	0.737	Sewell Seam, Fayette County, W. Va.
	B	0.761	Dorothy Seam, Crown City, W. Va.
	C	0.790	No. 3 Seam, McDowell County, W. Va.
	D	0.818	No. 3 Seam, McDowell County, W. Va.
	E	1.028	Beckley Seam, Raleigh County, W. Va.
Illinois	F	0.661	Herrin No. 6 Seam, Perry County, Ill.
	G	0.667	Herrin No. 6 Seam, Franklin County, Ill
	H	0.722	Harrisburg No. 5 Seam, Saline County, Ill.
	I	0.740	Belleville No. 6 Seam, Macoupin County, Ill.
	J	0.852	Springfield No. 5 Seam, Fulton County, Ill.

effect on the performance of two different types of filters. Furthermore, during the early part of the filter research program it was discovered that rather wide variations might be encountered in the results for tests performed on identical filters using dust mixtures of the same specifications. Apparently some of these variations were caused by slight differences in the construction of the filters, but the major variations appeared to be caused by some differences in the components of the dust mixtures. In order to eliminate the effects of variations in the dust constituents throughout any series of tests, it was found necessary to obtain a sufficient amount of each type of dust at the beginning of the series to last throughout all of the tests involved. From this it was evident that the specifications for the synthetic test dust should be determined more rigidly in order to aid duplication, and that possibly the ratios of the constituents should be changed in order to simulate more nearly the actual dust in the air.

The previous investigations showed that in so far as filter performance was concerned there were two general classes of dusts; first, the light flocculent dusts represented by lampblack and carbon black; and second, the more granular dusts represented by the coal ashes. Studies of dust samples taken from the air show reasonable counterparts of these two synthetic constituents.

Before making an extended investigation of various types of carbon and coal ash many other types of dusts which have been suggested and used to some extent for air filter tests were considered. Some of the dusts investigated, together with the reasons for not accepting them as satisfactory are shown in Table 1. While all of the dusts listed in this table were unsatisfactory for the reasons given, it is possible that further investigation may produce a material which will have all of the requirements and be more satisfactory than any combination of carbon and ash. However, as no such material was apparent, an extended investigation was made into the properties of carbon and ash dusts.

Ash

There is a wide variation possible in the chemical and physical properties of coal ash, even though the coal has the same commercial designation. These differences may be inherent in the original coal or may be due to some element in the combustion process by which the ash is formed. In order to investigate the effect of some of these variations on filter performance, samples were selected of coal from five different West Virginia Pocahontas fields and five different Illinois coal fields. All of these coals were burned at uniform rates of combustion and one of the Pocahontas samples was also burned at different rates of combustion. The resulting ashes were used for tests on two different types of filters.

In making these tests the jolted density of the resulting ash was determined by placing the dust in a cylindrical container $1\frac{7}{8}$ in. high and $2\frac{3}{32}$ in. in diameter, and tapping to maximum density. The source of the sample and densities of the resulting ash when burned at a uniform, moderate rate are shown in Table 2 and the densities of the ash for slow burning as compared with fast burning are shown for Sample *C* in Table 7. The moderate rate of combustion was estimated as that corresponding to the average hand-fired domestic furnace rather than the average stoker-fired furnace.

The various test dusts as shown in Tables 2 and 7 were used for testing both Filters *B-2* and *C*. In these tests the face air velocity was maintained at 300 fpm and the ash was fed to the filter at the rate of 40 g (grams) per hour. Each test was continued until the pressure drop across the filter had risen to 0.15 in. of water. The results of these tests are shown in Tables 3 and 7. The complete graphical test results for Filter *B-2* on the Illinois coal ashes at a moderate rate of combustion are shown in Fig. 2. These curves are typical for all of the tests.

The test results of Table 3 indicate that for Filter *B-2* the arrestance values are in close agreement for the different densities of each coal with the exception of the Pocahontas coal, *E*, which has a very high density and shows a somewhat lower arrestance value for this filter. For Filter *C* the arrestance values are not as consistent on either type of ash, although the variations are not serious. No doubt there is some difference in the particle size for the different ash samples as well as differences in density which will affect the arrestance values. There is a rather definite linear relationship between dust density and filter life for both the Pocahontas and Illinois coal ashes for both filters, as indicated in the curves of Fig. 3.

Since different filters were used with the different samples of ash it is likely that there would be some variation in the characteristics of the filters even though they were of the same make. In order to determine the effect of this variation, tests were made on five different filters of each type using Pocahontas ash from the same lot. The results for these tests as shown in Table 4 indicate a moderate variation between the different filters of the same type.

The figures of Table 5 give the variations in performance characteristics for Filters *B-2* and *C* for ash densities between 0.65 and 0.85 g per cubic centimeter.

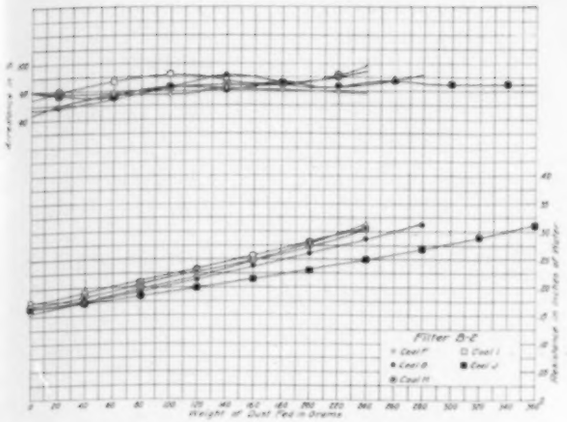

Fig. 2—Effect of five different Illinois coal ashes on performance of Filter *B-2*

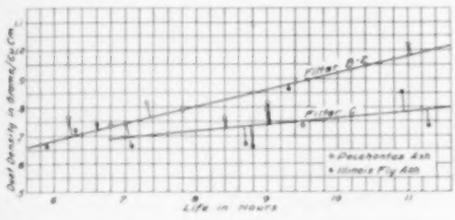

Fig. 3—Effect of ash dust density on filter life. Dust feed, 40 g per hour

As will be noted the range in arrestance values is very small. For Filter *B-2* the percentage variation from the average for arrestance, life and dust holding capacity are substantially the same when considering either the Pocahontas ashes or the Illinois ashes as a group, or the single Pocahontas ash. For Filter *C* the variations are greater for the group than for the single ash.

An indication of the effect of ash variation alone on filter performance is given in Table 6. The values shown in this table were obtained from those in Table 5 by subtracting the range in performance units for one ash from the range in performance units for various ashes and correcting the percentages accordingly. The results of this table are not all based on the same number of tests. The negative values may be considered within the range of experimental error.

The values in Tables 5 and 6 give an indication of the magnitude of the error which is likely to be introduced by the use of various ashes. They do not indicate true ranges of percentages as they are not based upon a sufficient number of tests for statistical accuracy.

Table 3—Filter Performance[a] for West Virginia Pocahontas and Illinois Coal Ashes on Filters *B-2* and *C*

(Dust feed 40 g per hour. Face air velocity 300 fpm.)

Source	Coal Sample	Jolted Density Grams/cc	Filter *B-2*			Filter *C*		
			Arrestance in Per Cent	Life in Hours	Dust Holding Capacity, Grams	Arrestance in Per Cent	Life in Hours	Dust Holding Capacity, Grams
West Virginia	E	1.028	90.8	11.0	400.1			
	D	0.818	94.2	7.3	275.1	85.5	9.0	307.7
	C	0.790	94.8	7.8	295.3	83.7	9.0	301.2
	B	0.761	93.2	6.2	233.3	78.3	8.4	262.5
	A	0.737	94.8	6.8	258.0	82.4	9.5	313.1
Illinois	J	0.852	95.4	9.3	355.4	82.9	10.9	362.0
	I	0.740	96.6	6.6	255.4	86.7	11.3	394.9
	H	0.722	96.0	6.3	241.9			
	G	0.667	95.6	7.1	271.0	84.6	8.7	294.9
	F	0.661	94.4	5.9	222.7	83.4	8.8	293.6

[a]Based on resistance rise of 0.15 in. of water across filter.

Table 4—Filter Performance[a] Variation for Filters *B-2* and *C* with Same Lot of Pocahontas Ash

(Dust feed 40 g per hour. Face air velocity 300 fpm)

Filter	Test No.	Performance Characteristics		
		Arrestance in Per Cent	Life in Hours	Dust Holding Capacity, Grams
B-2	1	93.2	7.0	261.2
	2	93.7	6.7	251.3
	3	94.8	7.8	295.3
	4	93.2	6.5	242.1
	5	93.2	7.0	261.2
C	1	82.1	9.7	319.6
	2	83.6	8.9	298.5
	3	83.7	9.0	301.2
	4	83.7	9.1	305.4
	5	81.7	9.2	299.6

[a]Based on resistance rise of 0.15 in. of water across filter.

The running of a sufficient number of tests to accomplish this would have involved too much time and expense and probably would have added little to the results. The tabulations in Table 6 indicate the greatest variations in performance characteristics to be for Filter *C*. This probably is due to the fact that variations in the average dust particle size have more effect on Filter *C* than on any of the other filters tested.

The test results for Filters *B-2* and *C* on the ashes from one Pocahontas coal burned at two different combustion rates are shown in Table 7. The slow rate of combustion was comparable with that of a well banked domestic fire, while the fast rate of combustion was somewhat in excess of that encountered in domestic installations. The ash from the fast burned coal had the highest density and, on filters, resulted in higher arrestance, longer life, and greater dust holding capacity than the slow burned ash. In the fast burned coal the lighter fractions of the ash probably were carried off with the flue gases, thus giving a denser ash of greater particle size. The greater average particle size would explain the higher arrestance, and the higher density the increased life and dust holding capacity. These tests indicate that any standards set up for the ash constituent of a dust mixture should include a limiting maximum combustion rate.

Carbon

In a previous investigation carried out in the general filter program four types of carbons (lampblack, carbon black, bone black, and soot) were selected and a study made of the effects of these materials on the performance characteristics of Filters *A*, *B-1*, *C*, and *D*. The results of this investigation showed that a wide range of results could be obtained dependent upon the particular carbon used, and that either lampblack or carbon black would make a satisfactory carbon ingredient for a standard dust if it could be duplicated.

In order to determine the variations in test results which might be expected from various carbon materials which would come within the code specifications, samples were selected of both lampblack and carbon black. (*Note*: the distinction between the two types of

Table 5—Filter Performance[a] Variation with Different Ashes
(Ash Density 0.65 to 0.85 g per cc)

Filter	Characteristics	Range for Various Pocahontas Ash		Range for Various Illinois Ash		Range for One Pocahontas Ash	
		Range in Performance Units	Range / Avg x 100 =	Range in Performance Units	Range / Avg x 100 =	Range in Performance Units	Range / Avg x 100 =
B-2	Arrestance	1.6%	1.7%	2.0%	2.1%	1.6%	1.7%
	Life	1.6 hours	22.9%	1.2 hours	18.5%	1.3 hours	18.6%
	Dust Holding Capacity	69.5 g	26.0%	48.3 g	19.5%	53.2 g	20.5%
C	Arrestance	7.2%	8.8%	3.3%	3.9%	2.0%	2.4%
	Life	1.1 hours	12.3%	2.6 hours	27.1%	0.8 hours	8.7%
	Dust Holding Capacity	50.6 g	17.1%	101.3 g	30.9%	21.1 g	6.9%

[a]Based on 0.15 in. resistance rise; face air velocity 300 fpm; dust feed 40 g per hour.

Table 6—Filter Performance[a] Variation with Different Ashes Corrected for Filter Variation
(Ash density 0.65 to 0.85 g per cc)

Filter	Performance Characteristic	Range for Various Pocahontas Ashes		Range for Various Illinois Ashes	
		Range in Performance Units	Range / Avg x 100 =	Range in Performance Units	Range / Avg x 100 =
B-2	Arrestance	0.0%	0.0%	0.4%	0.4%
	Life	0.3 hours	4.3%	—0.1 hours	— 1.5%
	Dust Holding Capacity	16.3 g	6.1%	—4.9 g	— 1.9%
C	Arrestance	5.2%	6.4%	1.3%	1.5%
	Life	0.3 hours	3.4%	1.8 hours	18.8%
	Dust Holding Capacity	29.5 g	10.0%	80.2 g	24.4%

[a]Based on 0.15 in. of water resistance rise; face air velocity 300 fpm; dust feed 40 g per hour.

Table 7—Effect of Ash Burning Rate on Performance[a] of Filters *B-2* and *C*
(Face air velocity 300 fpm. Dust feed 40 g per hour)

Filter	Ash	Density (Jolted) in g per cc	Arrestance Per Cent	Life in Hours	Dust Holding Capacity, g
B-2	No. 3 McDowell City Pocahontas (fast burned)	0.860	94.3	7.5	282.8
B-2	No. 3 McDowell City Pocahontas (slow burned)	0.765	90.2	5.5	198.2
C	No. 3 McDowell City Pocahontas (fast burned)	0.860	85.2	10.7	365.0
C	No. 3 McDowell City Pocahontas (slow burned)	0.765	78.9	7.9	249.5

[a]Based on 0.15 in. of water resistance rise; face air velocity 300 fpm; dust feed 40 g per hour.

Table 8—Comparison of Lampblacks[a] on Filter *B-1*
(Face air velocity 300 fpm. Dust feed 40 g per hour)

Source	Dust Sample	Dust Density	Avg Resis.,[b] Rise per Hour	Life[c] in Hours	Avg Arres. for 2 Hours
A	1	0.156	0.024	Approx. 4.5	28.1
	2	0.238	0.066	2.3	46.7
B	1	0.183	0.088	1.8	78.8
	2	0.213	0.089	1.8	74.0
	3	0.203	0.084	1.9	78.4
	4	0.206	0.063	2.3	75.6
	5	Approx. same as 1, 2	0.087	1.9	82.4
	6	Approx. same as 1, 2	0.084	1.9	64.4
	7	Approx. same as 1, 2	—	—	56.8
	8	Approx. same as 1, 2	0.098	1.8	66.2
C	1	0.209	0.074	2.1	71.1
D	1	0.265	0.049	1.5	58.4

[a]All lampblacks same brand and same manufacturer but from different supply houses.
[b]Based on 2 hour average.
[c]Based on final resistance of 0.30 in. of water across filter.

materials is that lampblack is formed by burning of oil, and carbon black by burning of gas.) All of the samples of each of these materials were of the same brand but selected from different lots and from material manufactured at different dates.

The test results from the different samples of lampblack are shown in Table 8. In this series of tests the face air velocity was maintained at 300 fpm with a dust feed of 40 g per hour. The same types of filters, *B-1*, were used in all tests. As it will be noted from the test results, there was a spread in arrestance values from 28.1 to 82.4. The lowest arrestance values were from lot *A*, and the 28 per cent arrestance was obtained on one of the earlier samples selected. There was a spread in the arrestance values for the samples of lot *B* from 56.8 to 82.4, and a spread in the values for life of filter from 1.5 to 4.5 hours. The significant point in these test results is the wide spread obtained in arrestance values and the uncertainty of consistent results from different samples of this material. It would be impractical to use a material giving such a wide range of test results as a basis for a synthetic test dust.

In making the tests on carbon black, two different brands of material were selected. These were known as Double-Bolted Carbon Dust and Super-spectra Carbon Black, both obtained from Binney and Smith Co., New York City. Six samples were selected from the first material and three from the second, all taken from different shipments manufactured at different times. In making these tests Filter *B-2* was used, the face velocity was maintained at 300 fpm and the dust feed at 20 g per hour. The rate of dust feed was set at 20 g as against 40 g for the lampblack because of the lower density of the carbon black. The test results on both brands of carbon black as given in Table 9 show a much higher degree of uniformity than those for lampblack. In the case of Super-spectra Carbon Black there was a marked drop in arrestance after the second hour of the test, which was probably caused by the inability of the oil to absorb the dust as rapidly as it was fed. Even though the rate was only 20 g per hour the extremely low density of this dust resulted in a high volumetric rate of feed. Of the various carbon dusts tested the Double-Bolted Carbon Dust appears to have the best properties for a dust to be selected as a standard to fulfill the carbon requirements in the code.

It should be noted that the requirements of a carbon dust for filter testing are very exacting and may be entirely different than the requirements of the dust when used for commercial purposes.

Dust Mixture

From an analysis of dust samples taken from the air it appears that there are two typical types of dust, and from a study of many types of artificial dusts it appears

that these two natural dusts in the air may be simulated with carbon black and Pocahontas ash, each prepared by a definite process. Tests have shown the performance characteristics of the various filters to be quite different when using the two different types of dust. Thus it appears that it is not necessary to use more than two different types of dust in a test mixture, but that it is important that these dusts be mixed in substantially the same proportions as they will be found in the air which must be handled by the filter.

Table 10—Volumetric Analysis of Dust Mixtures by Weight

Dust Mixture Per Cent by Weight	Vol. in cc of 40 g of Mixture	Vol. Per Cent Lampblack	Vol. Per Cent Carbon Dust	Vol. Per Cent Pocahontas Ash	Vol. Per Cent Illinois Ash	Vol. Per Cent Fuller's Earth	Vol. Per Cent Cottrell Ash
50 % Pocahontas Ash 20 % Illinois Ash 20 % Lampblack 10 % Fuller's Earth	95.0	54.0	.....	27.5	11.3	7.2	.....
50 % Pocahontas Ash 50 % Lampblack	154.3	83.0	.....	17.0	.....	.....	.....
90 % Pocahontas Ash 10 % Carbon Dust	72.2	.....	35.0	65.0	.....	.....	.....
80 % Pocahontas Ash 20 % Carbon Dust	92.4	.....	54.8	45.2	.....	.....	.....
70 % Pocahontas Ash 30 % Carbon Dust	112.5	.....	67.5	32.5	.....	.....	.....
100 % Cottrell Ash	31.1	.....	.....	.....	.....	.....	100.0

Most of the test dust mixtures used at the present time are proportioned on a weight basis. Since there is a wide variation in the densities of the individual dusts, there will be a big difference in the ratio of the components when calculated on a volume basis. The figures in Table 10 show the analysis of six different test dusts both on the weight and volumetric basis. The figures in the second column show the wide variation in densities of the different mixtures, and the figures for the percentages by weight and volume of the different dusts in the mixture show the wide variation in the components on the weight and volume basis. The sample made up of 50 per cent Pocahontas ash and 50 per cent lampblack as specified by the standard test code shows only 17 per cent of Pocahontas ash by volume and 83 per cent of lampblack. The density of the combined sample is less than one-third that of Cottrell ash, which is sometimes used as a test dust.

Although air-borne dust varies greatly in composition, depending upon such factors as locality, weather, time of year, etc., it seems improbable that it would ordinarily consist of more than 50 per cent by volume of carbon. This fact was recognized by manufacturers and users of filters when a mixture of 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack, and 10 per cent Fuller's Earth was selected as a basis for railroad air filter tests made at the University of Minnesota in 1937. Usually lampblack or carbon black is a very difficult dust to separate out of the air, and most filters which will successfully separate it show short life and low dust holding capacity. If a test dust mixture is to contain 83 per cent by volume of a dust which has very decisive effects on filter performance and which is found in much lower percentages in average air, it is evident that the rating of the filter will be misleading.

Table 9—Comparison of Filter *B-2* Performance Characteristics on Two Carbon Blacks

(Face air velocity 300 fpm. Dust feed 20 g per hour)

Brand Carbon Black	Sample No.	Density in g per cc (Jolted)	Resistance Rise in 3 Hours (In. of H_2O)	Avg Arrestance
Double Bolted Carbon Dust	1	0.154	0.118	52.5
	2	0.148	0.125	53.2
	3	0.162	0.130	59.6
	4	0.155	0.135	51.0
	5	0.171	0.123	54.7
	6	0.159	0.126	53.1
	7	0.177	0.143	59.2
Super-Spectra	1	0.066	0.152	64.5
	2	0.068	0.129	68.0
	3	0.068	0.131	67.7

It might be advisable to formulate more than one standard dust mixture for the purpose of rating filters to meet the requirements of the dust load which they must handle. Dust filters to be operated in smoky areas would be tested with a dust mixture incorporating a higher percentage of carbon than filters to be operated in areas relatively free from smoke. For the determination of the general performance of filters it would appear advisable to have a basic standard mixture, the alternate mixtures to be used only in case the manufacturer or purchaser desires ratings for specific purposes. The base mixture would be as representative as possible of average air-borne dust and result in a reasonable range in performance characteristics for the different filters. A dust mixture which shows a high arrestance on the average filter shows little range between the arrestance values for different filters. The opposite is true of dust resulting in a comparatively low arrestance on average filters. Furthermore, for the same weight of dust feed per hour, a low density dust will result in short filter life and low dust holding capacities, while a higher density dust will show the other extreme. A basic standard dust mixture should show none of these extremes.

Table 11—Comparison of Filter *B-2* Performance Characteristics[a] on Three Pocahontas Ash-Carbon Dust Mixtures

(Face air velocity 300 fpm. Rate of dust feed 40 g per hour)

Mixture	Avg Arrestance, Per Cent	Life in Hours	Dust Holding Capacity, g
70 % Pocahontas Ash 30 % Carbon Dust	84.4	3.5	117.4
80 % Pocahontas Ash 20 % Carbon Dust	87.8	4.0	140.7
90 % Pocahontas Ash 10 % Carbon Dust	90.9	5.0	182.0

[a]Based on final resistance of 0.30 in. of water across filter.

As previously explained, the investigations made on the various types of dusts indicate that the required elements of a dust mixture can be supplied by the two dusts, Pocahontas ash and carbon black. These two types of dust were thus selected and mixed in different ratios in order to find a suitable mixture as a basic test dust. The dusts were mixed in percentages by weight of 90-10, 80-20, 70-30 of Pocahontas ash and carbon dust respectively. These dust mixtures were used with Filter *B-2* at a face velocity of 300 fpm and a rate of

Table 12—Comparison of Filter Performance[a] on Four Test Dusts
(*Face air velocity* 300 *fpm. Dust feed* 40 *g per hour*)

Type Filter	50-50 Mixture			50-20-20-10 Mixture			80-20 Mixture			Cottrell		
	Arr.	Life	D.H.C.	Arr.	Life	D.H.C.	Arr.	Life	D.H.C.	Arr.	Life	D.H.C.
Filter A...	60.9%	9.0	219.1	79.9%	12.0	383.2	79.1%	10.0	316.3	92.3%	36.0	1329.0
Filter B-2..	.....	.....	.....	84.7%	3.0	101.7	87.8%	4.0	140.7	.....	.....	.....
Filter C...	59.0%	3.9	91.45	73.6%	6.0	175.0	70.1%	6.8	191.0	78.4%	22.8	715.0
Filter D...	92.7%	2.2	81.6	96.2%	4.9	188.7	94.4%	6.9	260.5	88.1%	14.0	1198.0

[a]Based on final resistance of 0.30 in. of water across filter.

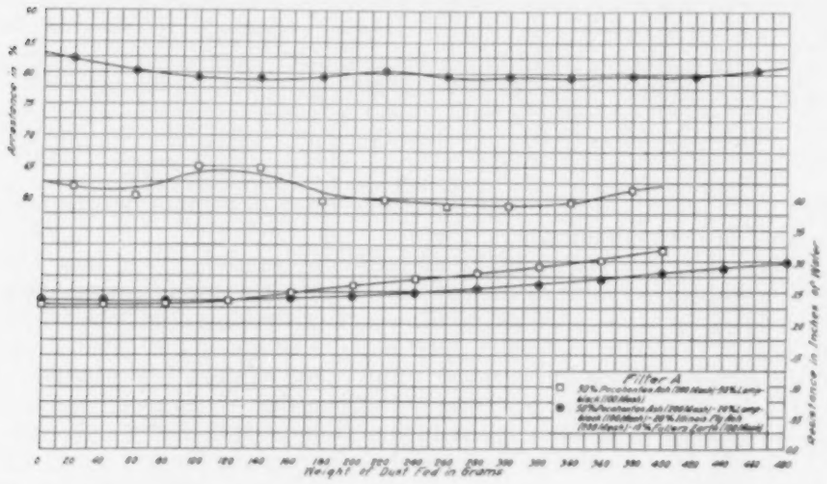

Fig. 4—Performance curves for Filter *A* on two dust mixtures. Dust feed, 40 g per hour

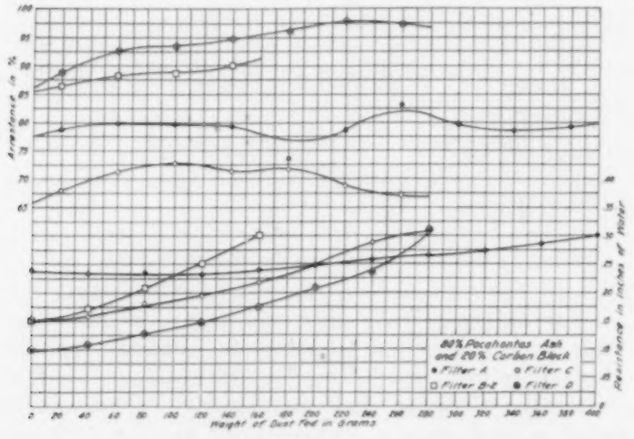

Fig. 5—Performance curves for Filters *A*, *B-2*, *C*, and *D* on 80-20 dust mixture. Dust feed, 40 g per hour

dust feed of 40 g per hour. From the results of these tests shown in Table 11 it was decided that the 80-20 ratio, giving approximately 50 per cent by volume of each type of dust, was the most representative of the three mixtures for a base test dust.

The 80-20 dust mixture was used for testing Filters *A*, *B-2*, *C*, and *D*, and the resulting characteristics compared with test results for the 50-50, 50-20-20-10, and the 100 per cent Cottrell ash tests. The results of these tests are shown in Table 12. A typical set of performance curves for the 50-50, and 50-20-20-10 mixtures is shown in Fig. 4 for Filter *A*, and the performance characteristics of the 80-20 mixture for Filters *A*, *B-2*, *C*, and *D* are shown in Fig. 5. The 50-50 mixture resulted in the lowest arrestance, shortest life and dust holding capacity, while the Cottrell ash showed the other extremes. The 50-20-20-10 mixture gave approximately the same results as the 80-20 mixture, probably because each contained 20 per cent by weight of carbon, and because Fuller's Earth and Pocahontas ash both have about the same effect on filter performance. The test values on these two mixtures lay between those for the 50-50 and the Cottrell test dusts.

A test dust such as Cottrell ash results in performance characteristics which are probably better than those actually obtained in practice, whereas a dust such as the 50-50 mixture probably penalizes the filter unduly. The 80-20 and the 50-20-20-10 mixtures give approximately the same results, but the former is easier to prepare because it has only two components.

Conclusions

There are two general classes of dusts in so far as their effects on the rating of air filters on a weight basis are concerned; first, the light flocculent dusts represented by lampblack and carbon black; and second, the more granular dusts represented by the coal ashes. These dusts vary in density and particle size and it is necessary to control these variations within reasonable limits if the effects of the dusts on filter performance are to be capable of duplication. Numerous tests conducted with various ashes and carbons indicate that carbon black and Pocahontas ash satisfy the requirements of a standard test dust if controlled within reasonable limits as to density and rate of burning. Changes in manufacturing processes may affect the standardization of the carbon black, but for any given set of manufacturing conditions there was less variation in the carbon blacks than the lampblacks tested. Fifty per cent by weight of lampblack as specified in the American Society of Heating and Ventilating Engineers Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work results in 83 per cent by volume of lampblack. This is apparently in excess of the amount of carbon ordinarily found in the air, and 20 per cent by weight of carbon appears to be a more reasonable specification. For these reasons the mixture of 80 per cent by weight of Pocahontas ash screened through a 200 mesh sieve and having a jolted density between 0.65 and 0.85 g per cubic centimeter, mixed with 20 per cent by weight of Double Bolted Carbon Dust screened through a 100 mesh screen and having a jolted density between 0.145 and 0.180 g per cubic centimeter is recommended as a standard test dust to be used for the general rating of air filters on a weight basis in the laboratory.

Cooling Tower Performance Studies

By L. M. K. Boelter,* Berkeley, Calif.

This report (Part 1) is the result of cooperative research sponsored by the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at the University of California, Berkeley, Calif.

Introduction

THIS report presents data indicative of the direction in which the Evaporative Cooling Project (cooling towers) is proceeding at the University of California as outlined by the ASHVE Technical Advisory Committee on Cooling Towers, Evaporative Condensers and Spray Ponds: B. M. Woods, Chairman; J. C. Albright, S. C. Coey, E. H. Hyde, E. H. Kendall, S. R. Lewis J. Lichtenstein, J. F. Park and E. H. Taze. Further instrumentation is required and is in progress. The tentative results on drop dynamics will be augmented by the application of these results to the mass and heat transfer phenomena. These results will be available by December 1939 and their application to the spray tower will be undertaken. Computations on the effect of drop size on vapor pressure, small drop thermal stability, liquid interface cooling due to evaporation and vapor pressure reduction due to solvents (impurities) are under consideration. Tests on the atmospheric towers are in progress during this summer (1939).

Description of Cooling Towers

The cooling towers available for test are shown in Figs. 1 to 6 and will be described briefly:

1. Cooling Tower No. 1: Atmospheric type model of multiple-section cooling tower with general characteristics as follows: Height 11 ft; width 4 ft; length 9 ft over-all (composed of two 4-ft sections).

Nine decks plus the distribution deck are spaced 1 ft apart. Distribution is accomplished by means of a central longitudinally-placed trough with small v-notch weirs along the upper edges.

The slats are $\frac{3}{8}$ x $3\frac{5}{8}$ in. x 3 ft 8 in., laid with their length parallel to the greatest dimension of the tower. The slat mats are arranged with six slats on the upper side and seven slats on the lower side across the width of the tower.

The louvers are at an angle of 60 deg with the horizontal. There are two louvers for each deck interspace.

2. Cooling Tower No. 2: Atmospheric type one-quarter full size scale model: Height 9 ft; length 6 ft; width 3 ft.

There are four aerator louver sections and a distributing section. Eleven decks plus the distribution deck are spaced at 8 in. Slats are $\frac{3}{4}$ x $\frac{3}{8}$ x 17 in. redwood.

Each deck is composed of eight 17 in. square sections. Each deck section contains 12 slats on the upper side and 13 slats on the lower side. Alternate decks are placed with the slats in opposite directions.

Distribution is accomplished by means of spray nozzles which discharge upward against special splash plates.

Approximate rating: 2 gpm ft² deck area; wind 3 mph; air at 70 F WB temperature; water cooling range = 10 deg; approach to wet-bulb is then approximately 8 F.

3. Cooling Tower No. 3: Forced Draft Spray Type. This tower is of sheet metal construction and pertinent data are as follows: Height 7 ft; cross-section 3 ft x 4 ft over-all (composed of two 2 ft x 3 ft chambers).

The design is such that the air first passes in parallel flow with the water downward through a 7 ft x 3 ft x 2 ft spray chamber and then passes upward in counter-flow with the water through a similar chamber.

There are six spray nozzles in each section.

The forced-draft fan is driven by a $\frac{1}{3}$ hp motor and is rated at 2000 cfm (actual capacity up to 3000 cfm).

Rating: Capacity = 3 tons; cooling range = 7 F.

4. Cooling Tower No. 4: Induced Draft Packed Type. This

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

Fig. 1—View of roof of Hesse Hall revealing louvers and spray pond

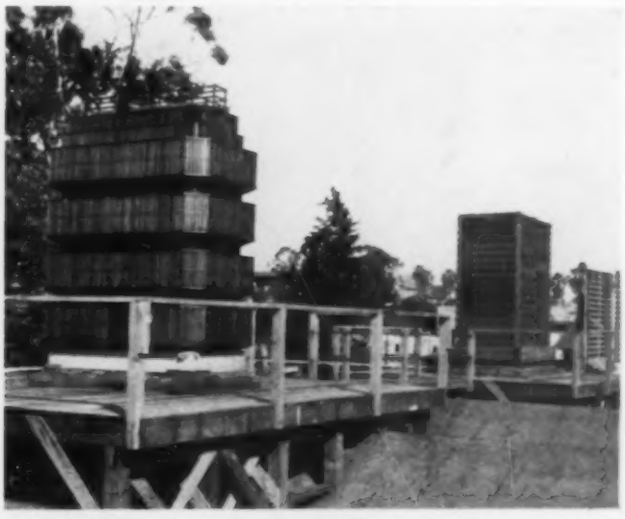

Fig. 2—View of towers No. 1 (*right*) and No. 2 (*left*) on roof of Hesse Hall

is an induced-draft counter-flow tower packed with special ovate wooden slats placed vertically. Pertinent data are as follows:

Height 12 ft, including a 3-ft drop-out chamber between the upper deck and the fan; cross section 3 ft x 3 ft. The casing is made of redwood shiplap, $\frac{13}{16}$ in. x 5 in. face.

There are 22 decks composed of ovate slats with 12 slats per deck and 13 slats per deck staggered in alternate layers. The ovate slats are 2¾ in. high by ⅞ in. maximum width.

Distribution is accomplished by means of a 3-in. I.D. wooden pipe across the center of the tower. This pipe runs transversely with respect to the direction of the slats, and holes are bored on the horizontal diameter directly above each distribution slat.

Performance data are available in the form of test results which include the following conditions: Air rate = 3 lb per second of dry air; water rate = 3.5 lb per second, or 3.06 gpm (square foot of cross section); dry-bulb temperature of entering air = 70 F; wet-bulb temperature of entering air = 60 F; water temperature in = 85 F; water temperature out = 75 F.

The roof of Hesse Hall is partially protected by louvers and an extension of the louver system is contemplated. The atmospheric towers have been placed on the roof of Hesse Hall (as shown in Figs. 1 and 2) while the forced-draft types were placed inside of the same building. The piping arrangement for all of the towers is interconnected and is diagrammed in Fig. 7.

In addition to the tower facilities a spray pond equipped with high and low pressure nozzles is available on the roof of Hesse Hall. Data have been taken on this spray pond but they have not yet been evaluated.

Description of Instruments Designed and Built

1. Hair Hygrometer. This instrument offers some advantages over other types for the determination of the space distribution of humidity. The design proposed by Büttner[1] is shown in Fig. 8 together with a typical calibration curve. A time lag inherent in the instrument precludes its utilization for atmospheric conditions in which the humidity varies rapidly with time. The variation of length of the human hair is amplified by an 0.01 in. diameter cantilever, the end deflection of which is measured by means of a microscope.

2. Thermocouple Psychrometer. Precision space psychrometry[2] demands a wick diameter which will provide adequate cooling to the wet bulb temperature at the local air velocity. Often the free convection generated by the wet junction depression is the only circulation available. Powell[3] and Cammerer[4] indicated the design limits of the thermocouple psychrometer. A unit designed for free convection circulation will possess a wick diameter less than 1 mm. In the type illustrated in Fig. 9, No. 36 constantan wire is plated with copper throughout part of its length thus providing two junctions. The length of the thermocouple wire must be sufficiently great to reduce conduction to the wet junction to a minimum. Either the wet junction and dry junction voltages or the wet junction depression and the dry junction voltages may be measured. The wick is either of dental floss or cotton properly boiled and is fed from a glass reservoir.

3. Absolute Hygrometer. In order that the water vapor content could be determined directly an absorption hygrometer (Fig. 10) was designed after Blackie[5]. The sample of moist air is drawn into the thermally insulated volumetric chamber A by reducing the mercury level to J by means of the leveling bottle M. Cock C is then opened to the sulfuric acid pipette F (the initial acid level being at E). Raising M then fills Chamber A with mercury and the air sample is displaced into the drying chamber E. Upon complete absorption of the moisture the acid level is restored to E, the mercury level to J, and the pressure on the manometer is read by means of the scale K. Upon correction for the air temperature change (indicated by thermometer B) the difference between the initial and final pressure readings will be the partial pressure of the water vapor.

4. Radiometer. In order to measure the rate of energy

[1]Büttner, K. Die Wärmeübertragung durch leitung und Konvection, Verdunstung, und Strahlung in Bioklimatologie und Meteorologie. *Abhandlungen, Prussian Meteorological Institute*, Vol. X, No. 5, p. 1-37 (1934).

[2]Walker, Lewis, McAdams and Gilliland, *Principles of Chemical Engineering*, p. 585, (1937).

[3]Powell, R. W., *Proc. Physical Society of London*, 48, 406-14, (1936).

[4]Cammerer, J. S., *Die Messtechnic*, Vol. XIII, No. 2, p. 21, (1937).

[5]Blackie, A., *Journal of Scientific Instruments*, 13, 6, (1936).

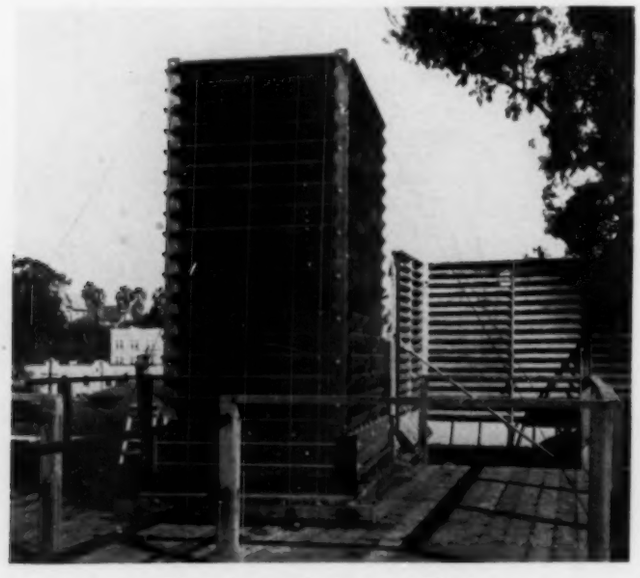

Fig. 3—Atmospheric tower No. 1

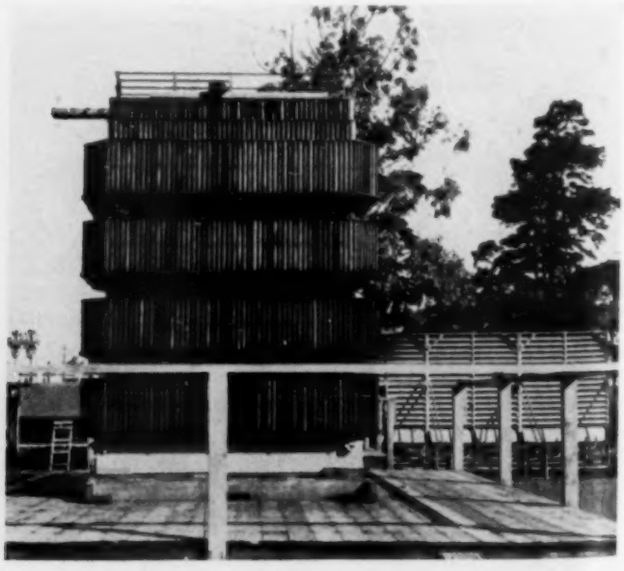

Fig. 4—Scale model atmospheric tower No. 2

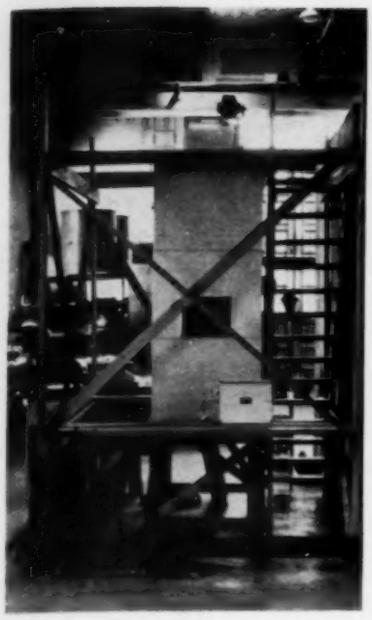

Fig. 5—Forced convection spray tower No. 3

radiated by the tower under operating conditions as well as to determine the rate of irradiation of the tower, a portable radiometer illustrated in Fig. 11 will be utilized. Briefly, the unit consists of a multijunction receiving unit placed at the apex of a conical aperture. The aperture is defined by baffles attached to a high heat capacity housing; blackened on the inside, chromium plated on the outside. The receiving element consists of 140 silver-constantan junctions, wound 104 per inch. The constantan is wound on a frame (thermal and electrical insulator) and one half of the winding is silver plated. The receiving junctions are covered with a *tin-foil* blackened strip. The inverse response (sensitivity) of this radiometer is 7.5 $\frac{\text{Btu}}{\text{foot}^2 \text{ hour mv}}$. The unit is frequently calibrated either by means of a hohlraum, a blackened surface at a known temperature or a thermal standard.

A pyrheliometer is available to measure solar and sky radiation.

5. Anemometer. A pressure plate anemometer[6] illustrated in Fig. 12 has been designed and partially constructed. The motion of a diaphragm, excited by the air stream, and resisted by a spring is transmitted to a portion of the core of a coil (transmitter) thus changing the reluctance of the magnetic circuit. A fixed reactor forms another leg of a bridge. Proper instrumentation will yield either the rms air velocity or its instantaneous value. The natural frequency of the moving parts is sufficiently high to measure the velocity fluctuations which will be effective in the cooling of a water stream.

Hot wire anemometers have also been designed for this work. A cup anemometer of special design yields the usual velocity data. The details of these units will be reported later.

6. Photography of Drops. The behavior of the drops within the tower will be studied stroboscopically. Special apparatus has been designed to photograph those particles which constitute drift. The apparatus shown in Fig. 13 is due to Gullberg. The particles are collected on a nitrate dope surface placed at *O*, and may be observed and photographed simultaneously. Photography may be accomplished under conditions of illumination which will not cause drop evaporation.

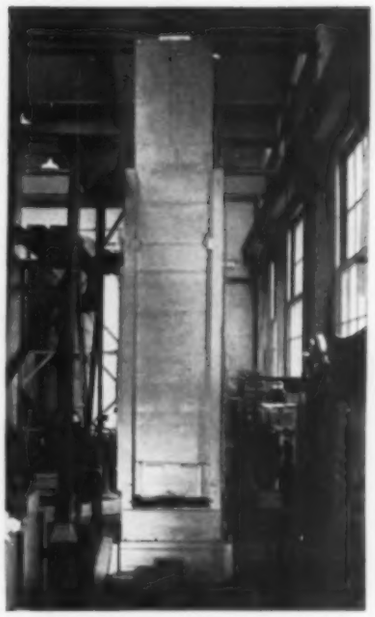

Fig. 6—Packed tower No. 4

Discussion of Operating Principles

The internal operation of a cooling tower is based entirely upon surface phenomena. The progressive disintegration of the liquid stream, which has been introduced into the tower, into many jets and small drops is governed by the principles of surface energy (surface tension), surface dynamics (capillary vibrations and considerations of stability), and the effect of the frictional drag of the air currents on the surface of the falling water. The liquid wets and spreads on the internal surfaces of the tower as a consequence of surface adsorption and the action of the interfacial surface tension; and once the surfaces are wet, a diffusion of some of the liquid across the surface and into the material (if porous) from

[6]Sherlock, R. H., and Stout, M. B., An Anemometer for a Study of Wind Gusts, *University of Michigan Eng. Research Bulletin* No. 20, May (1931).

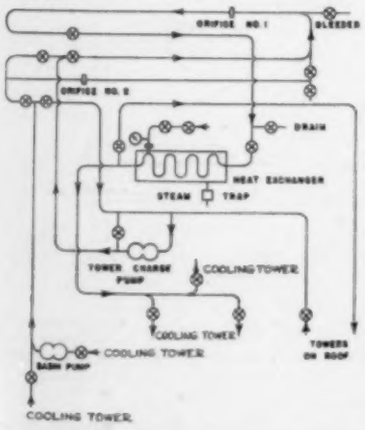

Fig. 7—Piping diagram for cooling tower liquid (water) distribution

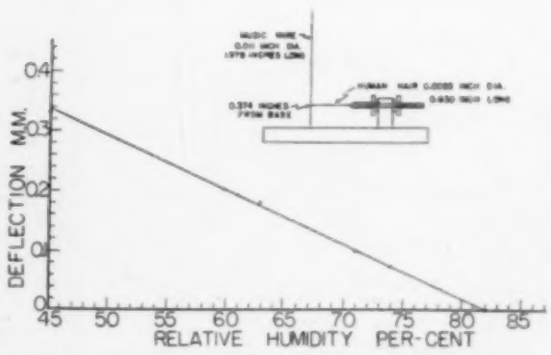

Fig. 8—Hair hygrometer after Büttner and the calibration thereof

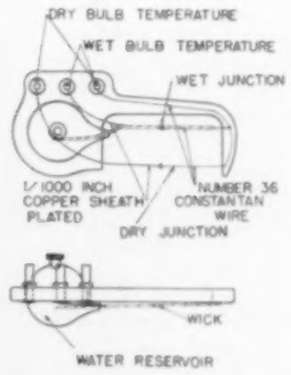

Fig. 9—Thermocouple psychrometer

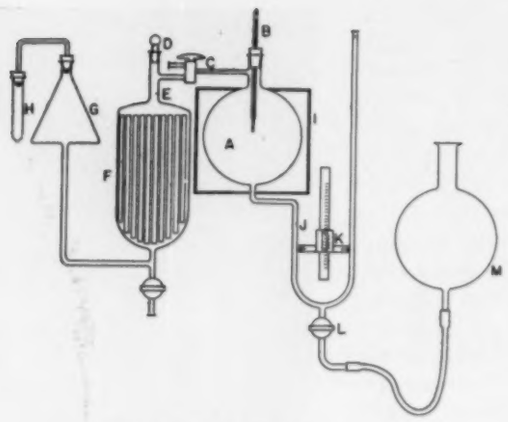

Fig. 10—Absolute hygrometer

A, Sampling chamber; *B*, Thermometer; *C*, Three-way cock; *D*, Stopper; *E*, Datum mark; *F*, Drying chamber; *G*, Reservoir; *H*, Calcium chloride tube; *I*, Aluminum foil screen; *J*, Datum mark; *K*, Sliding indicator; *L*, Cock; *M*, Mercury reservoir.

which the tower has been made takes place. The shattering and subsequent distribution of the jets and drops impinging upon the internal surfaces of the tower depend upon the microscopic characteristics of these surfaces and their arrangement.

The first phase of the evaporative cooling project will include a study of cooling tower behavior. Over-all performance data will be obtained for the several distinctive types of towers described earlier in this report. The behavior of these towers under different operating conditions will serve to reveal the different phases of the general problem which deserve particular experimental and analytical attention. The component unit processes which constitute the operation of a cooling tower will be outlined later. The end of the research project will be to present data and correlations which will make possible the rational design of a tower in terms of the independent variables. Some of these data are now available —others need to be determined. The data will be obtained and will apply to ideal systems, the composite of which will constitute the complete ideal tower. The actual tower performance should differ but little from the ideal tower performance if the basic design data are comprehensive and accurate.

When a cool unsaturated gas passes over the surface of a hot liquid, a simultaneous exchange of heat and material takes place by molecular mechanisms across the interphase boundary. Formulation of the resultant effect of these superimposed processes in terms of physical laws will lead to the only satisfactory methods of interpreting test results. The cooling tower may be considered as an extraction column in which one component is missing. The results of the test made on the towers will be presented in a form similar to those used to describe the behavior of absorption and extraction columns.

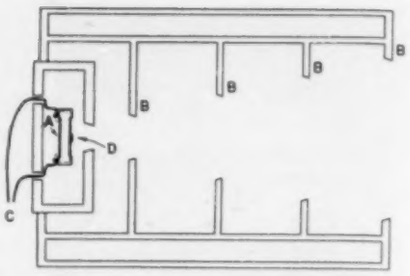

Fig. 11—Portable radiometer

A, Silver-constantan thermopile; *B*, Baffles; *C*, Lead-in-wires; *D*, Receiver.

The cooling process as it proceeds in the tower consists almost entirely of surface transfer phenomena. The liquid stream is progressively broken into jets, drops and sheets. In packed towers the liquid wets the internal surfaces, the resultant shattering of the drops, jets and sheets being dependent upon the surface characteristics of the packing. Evaporative cooling is directly dependent upon the phenomenon of surface diffusion, which involves considerations of surface configuration, rates of interfacial mass transfer, and surface concentrations and concentration gradients. Cooling by direct heat transfer, upon introduction of concepts of the rate of interfacial transfer of thermal energy by conduction and convection and surface radiation, is directly dependent upon the surface configuration, surface temperatures and temperature gradients at the surfaces, and the radiant emissivities of the surfaces. In addition, there are the perhaps inappreciable but nevertheless not-to-be-overlooked effects of the heats of surface adsorption, wetting and dilution, cooling effects which accompany the stretching of surface films, static electrical charges on the liquid and tower surfaces, changes in surface areas due to thermal expansion, chemical reactions which may take place on any of the boundary surfaces of the liquid-solid-gas system which exists within the cooling tower, and impurities which collect on the tower and liquid surfaces—(such as dust, oil, slime or solutes).

Comparatively little detailed infor-

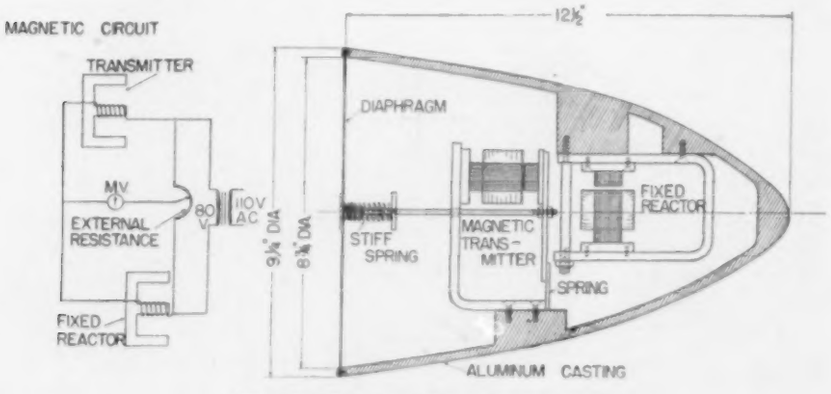

Fig. 12—Pressure plate anemometer

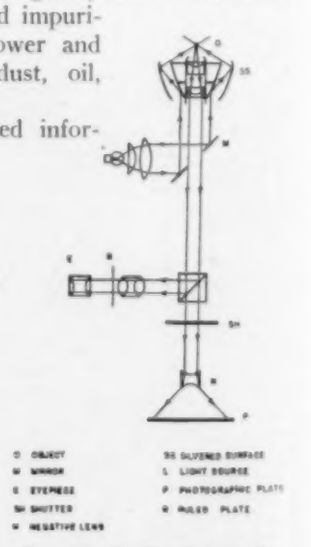

Fig. 13—Drift photography-arrangement of optical equipment

mation is available concerning phenomena of this type. And it is largely due to the lack of quantitative, (or qualitative in some cases), information concerning these microscopic details and their relation to the over-all performance of cooling towers under different operating conditions, that many uncertainties have arisen in connection with a complete understanding of the design and operation of present-day semi-empirically constructed towers.

Passing to a consideration of the relation between the internal operation and the many independent external variables, it is seen that for a given cooling duty of an atmospheric tower the constantly changing weather conditions and the rate at which cooling must be accomplished represent the independent variables to which operating conditions within a given tower must be adjusted.

In order to present an adequate formulation of the problem, it is necessary to assemble for consideration all of the variables which can affect the performance of a cooling tower in any way. In doing this, it would seem to be most significant to take a broad point of view and consider a *cooling tower* to be any structure which is operated for the purpose of cooling *any* liquid primarily by the method of evaporating a part of the liquid stream into a stream of gas as the two fluids flow through the structure.

Some of the variables noted here may be negligible in effect. In order to accomplish a fairly complete presentation, the list has not been restricted to the independent variables. Furthermore, all variables are to be considered in the light of different operating conditions in any given tower. The relative effects of the variables with different types of construction and with different fluids should be investigated together with the effects of the structural details themselves.

A possible grouping of three variables follows:

A. The Gas Circuit

1. The gas temperature and the variation throughout the structure.
2. The moisture content of the gas and the variation throughout the structure.
3. The barometric (total) pressure and the pressure variations throughout the structure.
4. The physical properties of the gas which have a bearing on tower performance and which commonly vary with the temperature, pressure and moisture content:—density, specific heat, thermal conductivity, viscosity, coefficient of thermal expansion, vapor diffusivity, apparent molecular weight and gas constant, number and type of gaseous ions present.
5. The rate of flow, over-all and in different sections of the tower.
   a. Velocity distribution and turbulence within the tower as affected by the type of construction, the rates of liquid and gas flow, and the weather conditions (initial conditions in the gas stream).
   b. Flow resistance characteristics of the tower passages and surfaces.
6. The weather variables. (The word *weather* as used here should be considered as a general term referring to the initial conditions in the gas stream. The relative importance of some of the variables in this category obviously depends largely upon the type of tower construction involved).
   a. Temperature and its variation
   b. Wind velocity and direction as periodic and transient functions of time
   c. Solar (nocturnal) radiation
   d. Humidity and its variation with related conditions
   e. Presence of dust and other impurities.

B. The Liquid Circuit

1. The liquid temperature and the manner of variation throughout the structure.
2. The physical properties of the liquid which have a bearing upon tower performance and which depend upon the liquid temperature: Density, specific heat, thermal conductivity, viscosity, vapor pressure, surface tension, latent heat of vaporization, coefficient of thermal expansion, surface emissivity—(Radiant energy exchange), molecular weight and gas constant of the vapor, vapor diffusivity.
3. The rate of flow and the size of the sheets, jets, and drop. (The cooling duty as related to the form and distribution of the liquid-gas surfaces of contact and the dynamical and thermodynamical conditions there existing.)
   a. Pressure and elevation head in the liquid supply system and details of the construction and installation of the supply and distribution system and its discharge characteristics.
   b. Liquid distribution within the tower as affected by the other variables. Degree of turbulence and the amount of free surface. Relative value of the different constructional details.
   c. The frictional drag of the gas on the liquid surfaces. Effect of fluctuations in velocity.
   d. Of prime importance are all conditions having a bearing upon the stability and dynamics of the liquid surfaces.
4. Adsorption of the liquid films by the tower surfaces.
   a. Microscopic surface characteristics. Interfacial surface tension.
   b. Effect of slime and solid deposits on the tower surfaces.
   c. Effect of paint on the tower surfaces.
   d. Diffusion of the liquid into the tower material.
5. Effect of dissolved gases and other impurities in the liquid upon the cooling process. (Modification of dynamical and thermodynamical conditions.)
6. Cooling or heating effects in the liquid circuit leading to or from the tower.

C. Charactertistics of the structure itself and the manner in which it is arranged and operated.

1. Constructional details.
   a. Dimensions, relative proportions.
   b. Materials.
   c. Arrangement and type of the internal surfaces.
   d. Louver design. Flow paths for gas stream. Relation to the drift loss.
   e. Liquid supply and distribution system.
   f. Levelness of all surfaces.
2. The temperature of the structure itself at different points as influenced by external and internal conditions.
3. Loss of liquid by entrainment. (Drift loss.) Effect of constructional details and weather conditions.
4. Design and location of the basin at the bottom of the tower. Surface evaporation from this basin.
5. Structural vibrations caused by the wind, the impact of the falling water, and extraneous effects,—*i.e.*, vibrations caused by nearby machinery or vibrations of the supporting structure if the tower is mounted on or in a building.
6. Noise generated by the air, water and structure vibration.
7. Location of the tower.
   a. Heat transfer to and from the surroundings,—especially by direct and reflected radiation.
   b. Orientation with respect to the prevailing wind velocity.
   c. Interference effects of surroundings.
   d. Drift loss as a nuisance.

8. Type of gas circulation provided for; *i.e.*, free or forced convection in any of the many special arrangements.
9. An important general item in this connection is a special study of the importance of the time and relative direction of contact of the gas and liquid and the interfacial surface areas and the resultant effects on tower performance as influenced by constructional details and arrangements and the operating conditions as determined by the many independent variables.

For example, the unit mass transfer conductance varies with the 0.37 power of the air velocity for air flow normal to the liquid, with the 0.56 power for transverse flow and the 0.80 power for longitudinal flow. Each type of relative motion is present in the atmospheric tower.

A second phase of the project will be to establish the model laws of cooling tower behavior. Upon presentation of these laws, tests on models may be extrapolated to the prototype with reasonable certainty. A third phase will be to attempt to formulate field testing technique and finally the fourth phase will be to study the economics of evaporative cooling.

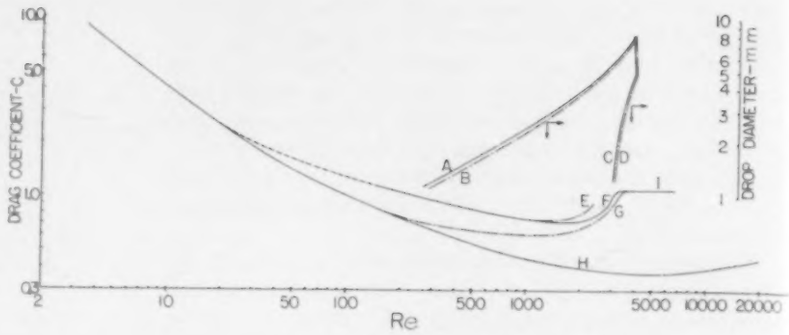

Fig. 15—Drop Dynamics

STEADY STATE REYNOLDS' NUMBER PLOTTED AS A FUNCTION OF DROP DIAMETER
A Flower and Wieselsberger
B Liznar and Wieselsberger

LIMITING REYNOLDS' NUMBER FOR STABLE DROP AS A FUNCTION OF DROP SIZE
C Lenard and Flower
D Lenard and Liznar

DRAG COEFFICIENT AS A FUNCTION OF REYNOLDS' NUMBER

E	Methyl salicylate drop	—Flower
F	Water drops	—Flower
G	Water drops	—Liznar
H	Spheres	—Wieselsberger
I	Flat plates	—Wieselsberger

Drop Dynamics

A study of the behavior of falling water drops will reveal results which may be applied to the problem of deck spacing in atmospheric (free convection) towers to the design of forced convection spray towers and to the design of cooling ponds. The knowledge of the dynamical behavior of drops will serve as the introductory step to the study of the heat and mass transfer experienced by drops as upon passage through the tower.

This behavior is most readily summarized in the form of velocity-distance-time curves for various drop sizes. These curves are computed in the following manner:

1. *The equation of motion. The equation determining the path of the drop when falling downward is*

$$mg - C\frac{A\rho v^2}{2} = m\frac{dv}{d\theta}$$

where m = mass of drop, v = velocity of drop, θ = time, $\frac{dv}{d\theta}$ = acceleration of drop, g = acceleration of gravity, ρ = mass density of the air (or any fluid through which the drop might fall), C = aerodynamic resistance coefficient defined by the equation $C = \frac{\text{Resistance offered by the air}}{\frac{1}{2}A\rho v^2}$, A = projected area of the drop normal to the direction of motion. Experimental data are available establishing C as a function of the Reynolds' Number for the drop, $Re = \frac{Vd}{\nu}$ where d = equivalent diameter of spherical drop of equal volume, ν = kinematic viscosity of the air.

A falling drop which encounters air resistance will be somewhat deformed compared to a true spherical shape. Photographs of falling drops have shown that a drop of diameter 1 or 2 mm or larger falling at the steady state velocity has the shape similar to that of an inverted acorn; *i.e.*, the leading surface is appreciably flattened while the trailing surface remains distinctly rounded but is slightly elongated.

This deformation gives rise to the question of how the projected area A and the equivalent diameter d should be measured. For want of a more satisfactory and convenient method, it has been the practice to refer these values to the equivalent spherical drop having the same volume as the actual drop. The satisfactory correlation of the aerodynamic resistance coefficient as a function of the Reynolds' Number on this basis justifies this method.

2. *Data on the steady-state velocity of falling drops and its application to the determination of the resistance coefficient as a function of Reynolds' Number.*

Liznar,[7] who summarized the data of Schmidt[8] and Lenard,[9] gives magnitudes of the steady-state velocity for falling water drops. Similar data are given by Flower.[10] Data on the increase of velocity over the steady-state velocity which is required to shatter a drop is reported by Lenard;[11] this information serves to establish the limiting velocity which is possible for a stable drop. For the air resistance of solid spheres, the data of Wieselsberger[12] is outstanding and has been re-checked

DROP DYNAMICS
STEADY STATE
VELOCITY-DROP
DIAMETER DATA

DROP DIA. MM	STEADY STATE VELOCITY FT/SEC		LIMITING VELOCITY FT/SEC	
	LIZNAR	FLOWER	LIZNAR	FLOWER
0.05	0.19	0.10		
0.2	2.57	2.37	600	730
0.5	6.25	7.02	260	295
1.0	10.83	12.75	147	152
2.0	17.13	19.20	79.7	81.0
3.0	21.0	23.85	57.0	58.9
4.0	25.0	26.03	46.0	47.0
5.0	26.4	26.9	39.5	40.0
6.0	25.8	26.6	32.4	32.4

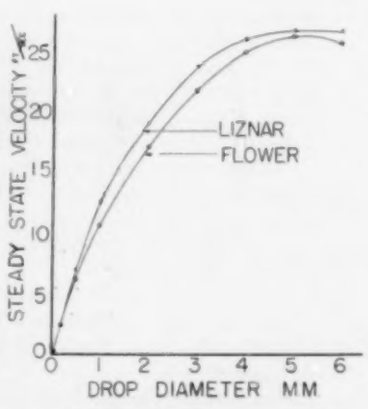

Fig. 14—Drop dynamics. Steady state velocity as a function of water drop diameter. Data of Liznar and Flower

[7]Liznar, J., *Meteorol. Zeitschr.*, 31, 339 (1914).
[8]Schmidt, W., *Akad. Wiss. Wien Sitzungber.*, 118:2a, 71 (1909).
[9]Lenard, P., *Meteorol. Zeitschr.*, 21, 249 (1904).
[10]Flower, W. D., *Proc. Phys. Soc. London*, 40, 167 (1928).
[11]Lenard, P., *Ann. Physik*, IV-65, 629 (1921).
[12]Wieselsberger, C., *Zeitschr. Flugtech. Motorluftschiffahrt*, 5, 140 (1914).

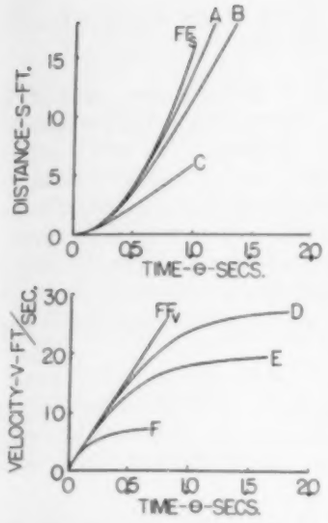

Fig. 16—Drop dynamics. Velocity-distance-time curves for water drops falling from rest through still air

A, D — diameter = 5.0 mm
B, E — diameter = 2.0 mm
C, F — diameter = 0.5 mm
FF — free fall

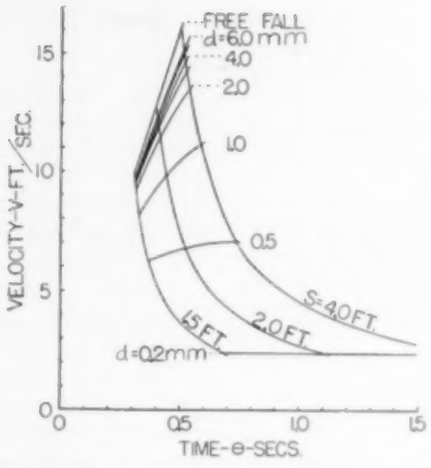

Fig. 17—Drop dynamics. The effect of the drop size upon the time required to fall distances of 1.5, 2.0 and 4.0 ft and the corresponding velocities attained. Drop falls from rest through still air

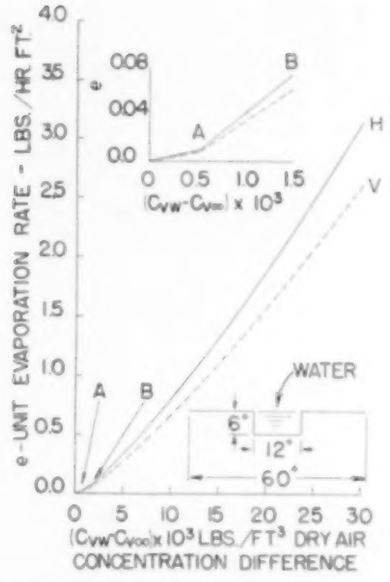

Fig. 18—Unit evaporation rate from quiet water surfaces to quiet air as a function of the vapor concentration difference at the liquid-air interface and far away

H — horizontal surface
V — vertical surface

Equations for *H*-surface:

O—A ..... $e=18.8\Delta c$ ..... $O<\Delta c<0.00052$

A—B ..... $e=-0.024+65\Delta c$ ..... $0.00052<\Delta c<0.0015$

B— ..... $e=251\ (\Delta c)^{1/3}$ ..... $0.0015<\Delta c<0.030$

by subsequent investigations of other workers.

The relation between the steady-state velocity and the drop diameter, revealed in Fig. 14, illustrates the manner in which the deformation of the larger drops gives rise to a greater increase in the resisting force of the air than the enlarged size increases the drop weight. According to the available data, it would appear that a drop of about 5 mm diameter falls with the maximum steady-state velocity. The extent to which the curve falls off for the larger drops is somewhat uncertain. Additional data are needed to completely establish the behavior of drops which have diameters of 6 and 7 mm.

Magnitudes of the limiting velocity for a stable drop, as determined by applying the velocity increments given by Lenard to the steady-state velocities are also given in the table of Fig. 14. These magnitudes may be utilized in the study of nozzle discharges and in determining the maximum possible size for a stable drop.

To determine magnitudes of the resistance coefficient, C, the steady-state velocity and the condition $\frac{dv}{d\theta}=O$ is substituted into the equation of motion which is then solved for C. Results of such computations are given in Fig. 15.

The effect of the deformation of the drop in increasing the resistance above that for a sphere at the higher magnitudes of Reynolds' Number is also shown here, the resistance coefficient increasing to the magnitude corresponding to a flat circular plate (it may even go above this value, but the available data are of a limited range and do not establish this point). The effect of the surface tension of the drop upon the degree of deformation under given flow conditions may be obtained from the data of Flower for methyl salicylate, which possesses a surface tension about one-half of that for water. Magnitudes of Reynolds' Number corresponding to the steady-state and limiting velocities are also shown as they depend upon the drop size; it is seen that the two curves intersect at a drop diameter of about 8 mm, indicating that no larger drop could be stable. Measurements of the size of raindrops[5] serve to confirm this point. In an ordinary heavy rain the largest drops are of the order of 5 to 6 mm in diameter. Heavy thunderstorms contain a few drops up to 6.5 or 7 mm in diameter.

The wide spread between the data of Flower and that of Liznar requires some explanation. At first it was thought that the divergence was due to a combination of different experimental methods and the effect of upstream turbulence, for Flower measured (photographically) the velocity of drops falling through still air, while Lenard measured the velocity of an air stream required to just balance the weight of the drop. However, subsequent computations yielding the distance-velocity curves indicate that Flower did not actually measure the steady-state velocity, for the computed distance of fall required to attain this velocity for the larger drops is about double the distance utilized in Flower's tests. It would seem that this difficulty arose because of Flower's use of a ballistic balance to determine the distance of fall necessary to give the steady-state velocity as manifested by a maximum in the throw of the balance upon the impact of a drop. But since the rate of change of velocity with distance becomes very small as the steady-state is approached, it is doubtful whether the true limit could be detected by a ballistic balance.

3. *Integration of the equation of motion.*

The form of the equation and the way in which C varies with Reynolds' Number make it necessary to resort to graphical integration. The velocity-time curve is obtained by evaluating

$$\int_0^\theta d\theta = \int_0^v \frac{dv}{g-C\frac{A\rho v^2}{2m}}$$

where the factor $\left(g-C\frac{A\rho v^2}{2m}\right)$ represents the acceleration per unit mass.

A second graphical integration yields the distance-time curve according to the relation

$$\int_0^s ds = \int_0^\theta v\,d\theta$$

The velocity-distance-time curves have been obtained in this manner for drops of diameter 0.2, 0.5, 1, 2, 3, 4, 5, and 6 mm,

both for the case in which the drop falls from rest and that in which it is discharged (either upward or downward) from a nozzle. Two sets of curves have been computed, one according to the data of Flower and one according to the data of Liznar. Even though it is believed that the magnitudes according to Liznar are the more accurate, the use of both sets of data will serve to indicate the effect of changes in the value of the resistance coefficient and the range of behavior which one may expect.

These curves have been computed for the case in which the drop falls through still air, but they may be applied to the case in which the air is in motion upon recognition of the fact that the velocity of the drop as given is that relative to the air through which the drop is passing.

4. *Velocity-distance-time curves for drops falling from rest.* Characteristic relationships can be brought out clearly for drops of 0.5, 2, and 5 mm diameter as representative magnitudes computed according to the *data of Liznar.*

Table 1

Drop Diameter Time Sec	Free Fall		d = 0.5 mm		d = 2.0 mm		d = 5.0 mm	
	v ft/sec	s ft	v ft/sec	s ft	v ft/sec	s ft	v ft/sec	s ft
0.1	3.18	0.16	2.85	0.145	3.15	0.15	3.18	0.16
0.2	.....	0.64	4.75	0.538	6.05	0.61	6.32	0.638
0.3	.....	1.45	5.84	1.078	8.82	1.37	9.94	1.43
0.4	.....	2.57	6.40	1.69	11.10	2.36	12.12	2.50
0.5	16.08	4.02	6.73	2.36	12.95	3.57	14.62	3.84
0.6	.....	5.79	6.94	3.04	14.47	4.93	16.91	5.42
0.68	.....	.....	7.02+	3.59+	.....	.....	.....	.....
0.7	.....	7.88	.....	.....	15.66	6.45	19.02	7.23
0.8	25.73	10.29	.....	4.43	16.59	8.08	20.85	9.22
0.9	.....	13.02	.....	.....	17.30	9.82	22.26	11.37
1.0	.....	16.08	.....	5.85	17.81	11.54	23.40	13.64
1.18	.....	.....	.....	.....	.....	.....	.....	18.00*
1.2	.....	.....	.....	.....	18.41	15.31	24.96	.....
1.356	.....	.....	.....	.....	.....	18.00*	.....	.....
1.4	.....	.....	.....	.....	18.84	.....	25.82	.....
1.6	.....	.....	.....	.....	19.15	.....	26.45	.....
1.66	.....	.....	.....	.....	19.20+	.....	.....	.....
1.84	.....	.....	.....	.....	.....	.....	26.9+	.....

*—limit of plot. Interpolation equations are used for greater distances.
+—steady state velocity.

Magnitudes are tabulated in Table 1 and are plotted in Fig. 16.

5. *The time required to travel a given distance and the velocity attained in this distance as related to the diameter of a drop falling from rest.*

Data of this type have immediate application to the atmospheric cooling tower and are tabulated in Table 2 as well as being plotted in Fig. 17.

Table 2—Data of Liznar

Distance of Fall	S = 1.5 ft		S = 2.5 ft		S = 4.0 ft	
Drop Diameter mm	θ sec.	v ft/sec	θ sec.	v ft/sec	θ sec.	v ft/sec
0.2	0.696	2.367	1.12	2.367	1.68	2.367
0.5	0.371	6.28	0.620	6.77	0.738	7.02
1.0	0.3275	8.10	0.439	9.80	0.582	11.12
2.0	0.315	9.18	0.412	11.34	0.533	13.50
3.0	0.311	9.50	0.403	11.87	0.516	14.17
4.0	0.310	9.56	0.402	12.00	0.514	14.66
5.0	0.309	9.59	0.401	12.15	0.512	14.90
6.0	0.308	9.60	0.400	12.16	0.510	15.05
Free fall	0.307	9.86	0.396	12.73	0.499	16.06

The data presented in Tables 1 and 2 on drop dynamics will serve as an introduction to a more comprehensive report which will be available within the year.

The following tasks are now being accomplished:

1. Correlation of the velocity-distance-time curves for drops discharged from a nozzle and decelerated by the air resistance.
2. Heat transfer computations for the evaporative cooling of falling drops.
3. Studies of the dispersion of jets and attempts to definitely evaluate the deformation of the drop as it encounters the resistance of the air.
4. Considerations relating to the dynamics of drop impacts and heat transfer accompanying such action.

Evaporation from a Quiet Water Surface Into Quiet Air

These experiments[13, 14, 15] furnish the data from which the *minimum* rates of evaporation from a horizontal water surface may be predicted. No water surface existing in and about cooling towers will be bounded by quiet air. Mass and heat transfer will then occur in excess of that rate which is due only to free convection of the air. Therefore the data presented in Fig. 18 may be considered as the minimum unit evaporation rate which will be experienced from water surfaces existing in cooling tower systems.

The data shown apply only for a 1-ft diameter pan, distilled water being held at the level of the surrounding 5x5 ft floor. The pan was completely housed in order to prevent cross flow due to thermal and dynamic wind forces.

The experimental data lie in a range in which the unit evaporation rates [pound per (square foot) (hour)] are not quite independent of pan diameter, varying approximately inversely as the quarter power of the diameter. As a first approximation the results may be used for other surface areas with diverse surroundings.

The analogy between heat transfer and mass transfer in free convection has been established for a limited range of the pertinent variables and for one geometrical arrangement. For heat transfer the conductance per unit area for the top of a horizontal surface is approximately 1.2 times that of a vertical surface[16]. Although based on meagre evidence and on the validity of the heat transfer—mass transfer analogy, the evaporation data obtained for a horizontal surface will be divided by 1.2 in order to obtain a first approximation to the mass transfer of vapor from a *stationary* vertical water sheet into quiet air at different humidities. These data are also shown in Fig. 18 and may be used to predict the minimum rate of evaporation from a vertical water sheet to quiet air.

Results on the Mechanically Induced Draft, Contra-Flow, Packed Cooling Tower

A photograph of this tower is shown in Fig. 6 and construction details are revealed in Fig. 19. A description of the tower is included earlier in the paper. The piping arrangement is shown in Fig. 7.

The tower was tested[17] at various air rates and at various water rates as indicated in Fig. 20. Mass and heat balances were accomplished for each run. The average discrepancy was approximately ±3.5 per cent. The pressure drop through the tower has not yet been measured. No attempt has been made to condition the entering air so that a complete series of runs for various humidities must be made throughout the period of a year (low humidities usually occur only in September). Entering air relative humidities ranged around 60 per cent, dry-bulb temperatures ranged from 65 F to 70 F.

[13] Sharpley, B. F., and Boelter, L. M. K., *Ind. Eng. Chem.*, 26, 1125-1131 (1938).

[14] Gordon, H. S. Thesis M. S. Degree 1938. Evaporation from a One-foot Diameter Water Surface into Quiet Air. (On file in the University of California Library, Berkeley.)

[15] Griffin, J. R. Thesis M. S. Degree 1939. Evaporation from a Horizontal Water Surface into Air by Free Convection. (On file in the University of California Library, Berkeley.)

[16] Bosch, M. ten, Die Wärmeübetragung, 3rd Ed., p. 167, Berlin, Julius Springer, (1936).

[17] London, A. L. Thesis M. S. Degree 1938. The Performance Characteristics of a Mechanically Induced Draft, Contra-Flow, Packed Cooling Tower. (On file in the University of California Library, Berkeley.)

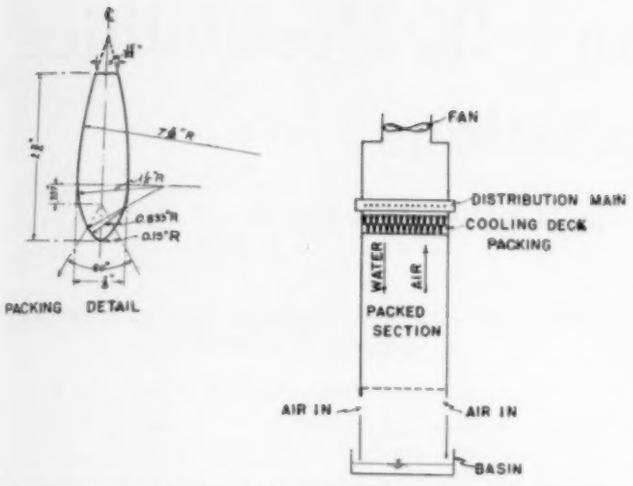

Fig. 19—Construction details of airfoil-section packed tower

and entering water temperatures ranged from 80 F to 103 F.

The effectiveness of the tower was defined in the following manner:

ϵ_H = enthalpy reduction of water divided by the maximum possible enthalpy decrease of the water if the exit air is saturated at the inlet water temperature. This effectiveness ratio defines the tower performance as an air heater and humidifier and is shown as a function of the air rate in Fig. 20. It will be noted that the effectiveness is independent of the water rate down to 1.75 lb per second, below which partial wetting of the packing results.

A dimensionless ratio,[18] named the number of Transfer Units, $\frac{f_D A}{G}$, *where* f_D = mass transfer conductance per unit area in pounds per hour foot^2(pound/pound), A = area of packing in foot^2 and G = air rate in pounds per hour may be used as the generalized variable instead of the air rate G. The actual length of the tower (l) divided by the number of transfer units is called the height of a transfer unit, HTU. The HTU for the packed tower ranges from 3.5 to 8.5 ft. The behavior of the tower can be expressed empirically by the equation

$$\epsilon_H = 1 - 1.40e^{\frac{-f_D A}{G}\eta}$$

where η = fraction of the packing which is wetted and e = Napierian base. $\eta = 1$ for water rates greater than 1.75 lb per second. Another effectiveness ratio which is based upon cooling performance may be defined as:

ϵ_c = enthalpy decrease of water divided by the maximum possible decrease, provided the water exit temperature coincides with the inlet air wet-bulb or

$$\epsilon_c = \frac{\textit{decrease of water temperature}}{\textit{inlet water temperature—inlet air wet-bulb temperature}}$$

The range of variation ϵ_H and ϵ_c is almost identical, although the variation is opposite. For instance, for the same liquid rate, the magnitude of ϵ_c for the high gas rate will correspond to that of ϵ_H for a low gas rate. The effectiveness as a cooling unit increases with increased gas rate.

A more complete analysis of the results is under consideration, which, when added to the pressure drop data, will be presented in a future paper.

Conclusions

1. Progress on the specific problem of evaporative cooling of water by direct contact with air is being made. The variables have been segregated and the effect of each independent variable will be studied separately. For instance, models of the full-scale atmospheric towers will be placed in a wind stream of known turbulence in order to study the air flow through them. Air-free water will be used to determine the effect of dissolved gases. Further progress will be made in the ensuing year.

2. The studies in drop dynamics and the related computations on the evaporative cooling of water drops to follow will make available design data of fundamental importance.

3. Several effectiveness ratios of cooling towers have been proposed based on the concept of an air heating device and of a water cooling device. Each has its place in design calculations.

[18] See reference 2, p. 490.

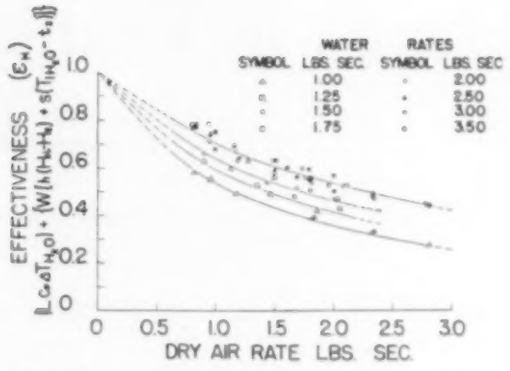

Fig. 20—The effectiveness of the airfoil-section packed tower as a function of the air rate for various water rates

$$\epsilon_H = 1 - 1.40e^{\frac{-\eta f_D A}{G}}$$

$\eta = 1.00$ $L \geq 1.75$ lb per second water rate
$\eta = 0.87$ $L = 1.50$ lb per second water rate
$\eta = 0.79$ $L = 1.25$ lb per second water rate
$\eta = 0.71$ $L = 1.00$ lb per second water rate

Acknowledgments

Technical and technological contributions were made by the following men, students, graduate students, faculty and mechanicians: T. F. Ascherfeld, H. E. Brown, V. H. Cherry, F. L. Dickey, H. L. Eagles, J. T. Gier, H. S. Gordon, J. R. Griffin, J. E. Gullberg, M. L. Hilleger, E. D. Howe, A. L. London, W. E. Mason, T. H. Morton, S. A. Murphy, Jr., H. B. Nottage, W. F. Pemberton, B. F. Raber, F. A. Ryder, H. J. Smith, and H. W. Sylvis.

The technical suggestions of the membership and committees of the Los Angeles and Golden Gate Chapters of the American Society of Heating and Ventilating Engineers and the *Air Conditioning Society of San Francisco* serve to guide the work. The interest and activities in behalf of the project by Prof. B. M. Woods are stimulating and effective. Construction and maintenance of the equipment is accomplished through the aid of WPA Mechanical Engineering Project No. 8850.

The assistance of California industry is also gratefully acknowledged.

Smoke Hazards of Air Conditioning Systems

Recent results of investigation by the Underwriters' Laboratories, Inc., requested by the National Board of Fire Underwriters, are discussed in a pamphlet issued by the Board, containing suggestions and recommendations for the reduction in loss of life and property from fire. It also covers tests and research on practitcal and effective means of detecting and controlling the spread of smoke in air conditioning systems and of reducing the possibility of smoke damage and panic hazard.

Ozone in Ventilation—Its Possibilities and Limitations

By William N. Witheridge* and Constantin P. Yaglou** (*MEMBER*)

THE purpose of this work was to study the possibilities and limitations of ozone for counteracting body odors in ordinary occupied rooms. Answers were sought to the following questions:

(a) How much ozone is required to counteract body odors under various practical conditions?

(b) How are the occupants affected by such ozone concentrations?

(c) Does ozone oxidize body odors in the air, or does it "mask" them under practical conditions of ventilation?

The action of ozone on various odors, such as hydrogen sulfide, butyric and valeric acids, indol, skatol, ammonia, stale urine, and others, has been studied by several investigators, but to our knowledge, there are no quantitative data on the effect of low ozone concentrations upon body odors in the air of ordinary occupied rooms. A serious obstacle to these studies has been the lack of a sensitive chemical method for measuring such low ozone concentrations as are permissible in ventilation work. Moreover, knowledge is still lacking on the constituents that make up the *body-odor complex*, and the only basis there is to work on is the smell of the complex and the quantity of clean air required to dilute it to the threshold concentration.

Low ozone concentrations in rooms estimated from the ozonator output and extent of dilution are open to criticism because of the great variability in decomposition of ozone depending on the characteristics of a room and of the air supply system, absolute humidity which appears to be very important, temperature, and possibly other factors that may arise under practical conditions. Body odors, too, have peculiar disappearance characteristics of their own, and it is difficult to predict their interaction with ozone except by actual tests using the human nose for judging the results.

In the present study, low ozone concentrations were estimated from the strength of smell, using a technic developed in previous works[1, 2, 3]. An interesting finding which permitted estimation of ozone concentration in the presence of body odors was the inability of the latter to affect the ozone smell. With proper technic, there was no difficulty in distinguishing ozone intensities regardless of body-odor strength.

Review of Physical, Physiological and Pharmacological Properties

Ozone consists of three atoms of oxygen in an unstable combination, represented as $O_3 \rightarrow O_2 + O$, and the oxygen atom set free possesses great oxidizing power. Ozone is a normal constituent of country and polar air in detectable, but ill-defined, concentrations. It is produced in nature photochemically by ultra-violet light, and is almost absent in city air, except during thunderstorms. Even in the absence of oxidizable material, the rate of decomposition of ozone is known to increase appreciably with a rise of humidity and to a smaller extent with temperature rise also.

Ozone has a pungent odor, somewhat similar to that of chlorine or garlic, and is, therefore, a very effective *masking* agent for odors which are not readily oxidized. In high concentrations it readily oxidizes organic matter and has a variety of applications in water sterilization, bleaching, and in the control of fungi and spores in cold-storage rooms. The oxidation of body odors by ozone at concentrations that are permissible in ventilation work is highly debatable. Erlandsen and Schwarz[4], Hill and Flack[5], Jordan and Carlson[6], Konrich[7], and others working on odors that are supposedly related to the body-odor complex, ascribed the counteraction to *masking*, rather than to oxidation. These studies were severely criticized by the ozone industry, directly or indirectly, but no authoritative data have been presented to refute the findings.

The toxicologic properties of ozone are fairly well known. Even in relatively low concentrations (0.1 to 1 ppm) it exerts a powerful irritating action on the mucous membranes of the respiratory organs and shortens the lives of guinea pigs after months of continuous exposures.[8] Pneumonia is the terminal condition. Much higher concentrations lead to death in shorter periods from lung congestion and edema[5 to 9]. There is no systemic poisoning, as most of the ozone is decomposed on the mucous membranes of the respiratory tract, where its destructive action is mainly localized.

The irritation threshold appears to be about 0.05 ppm; there is no evidence of damage to health in concentrations well below this. Workers in the London subways

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[1]Fair, G. M.: On the Determination of Odors and Tastes in Water. (*Journal New England Water Works Association*, 47: 248, 1933; a brief review of literature.)

[2]Yaglou, C. P., Riley, E. C., and Coggins, D. I.: Ventilation Requirements. (ASHVE TRANSACTIONS, 42: 133, 1936.)

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

[4]Erlandsen, A., and Schwarz, L.: Experimentelle Untersuchungen über Luftozonisierung. (*Zeitschr. f. Hyg. u. Infektions-Krankh.*, 67: 391, 1910.)

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[7]Konrich: Zur Verwendung des Ozons in der Lüftung. (*Zeitschr. f. Hyg.*, 73: 443, 1913.)

[8]McDonnell, H. B.: Experiments with Ozone on Guinea Pigs at the James Todd Laboratory of the University of Maryland. (*Journal Association Official Agricultural Chemists*, 13: 19, 1930. Also personal communication.)

[9]Sawyer, W. A., et al: The Alleged Purification of Air by the Ozone Machine. (*Journal American Medical Association*, 61: 1013, 1913.)

Table 1—Effects of Ozone in Various Concentrations

Parts Ozone per Million Parts of Air (by Vol.)	Observed Effects	Authority
0.01	Threshold for keen sense of smell.	Hill's Discussion (19) (Confirmed by present study)
0.015	Normal threshold: Readily detectable by most normal persons	Present study
0.04	Upper allowable limit in occupied rooms	Present study
0.10	Objectionable to all normal persons; irritating to mucous membranes of nose and throat in most persons	Present study
0.5–1.0	Disorders breathing, reduces oxygen consumption and shortens lives of guinea pigs	(5), (8), (13)
0.3–1.5	Inhibits fungus and mold growth in cold-storage rooms	(14), (15)
1–10	Headache, respiratory irritation, and possible coma	(5), (6)
15–20	Lethal in 2 hours to small animals	(5)
> 1,700	Lethal in a few minutes	(8)
6,500	Germicidal for air-borne organisms	(15)

have shown no ill effects after years of exposure to concentrations at or slightly above the olfactory threshold. In Table 1 are outlined the effects at various concentrations, as reported by different investigators.

It is now generally recognized that ozone is not a satisfactory germicide at concentrations within the range of human tolerance (0.01 to 0.04 ppm); its effectiveness varies with the concentration, the degree of atmospheric humidity, and last, but not least, with the species of bacteria involved[5, 6, 9, 10, 11].

On certain species, Jordan and Carlson[6] were unable to obtain any sure germicidal action with amounts of ozone from 3 to 4.6 ppm. Concentrations sufficient to kill dry typhus bacilli, staphylococci, or streptococci in the course of several hours killed guinea pigs sooner in Sawyer's experiments.[9]

Claims for converting carbon monoxide to carbon dioxide by the use of ozone in garages are now generally discredited.[12]

Method of Present Study

Description of Experimental Rooms: The experiments were carried out in the spring and summer of 1936. Two adjacent rooms were used, each having a net air space of about 1,400 cu ft. One of the rooms designated hereafter as the *experimental* room, was occupied by the subjects who produced the odor to be counteracted by ozone. The other room served as a *control* room for pure air and was occupied by not more than two *judges* at a time who estimated the odor intensities in the experimental room. A small tight door, closing against felt cushions, allowed passage of the judges from the control to the experimental room for estimating the ozone or body-odor intensity and quality of air. All window and door cracks in the experimental room were sealed with plasticene and taped over with adhesive tape. In tests with low airflows, the small connecting door was

[10]Wagner, R. F.: Experiments on Sterilization of Air. (A thesis to the Department of Sanitary Engineering, Harvard University, 1936.)
[11]Kendall, A. I., and Walker, A. W.: The Effects of Ozone Upon Certain Bacteria (Colon Bacilli and Staphylococci) and Their Respective Phages. (*Journal Infectious Diseases*, 58: 204, 1936.)
[12]Salls, C. M.: The Ozone Fallacy in Garage Ventilation. (*Journal Industrial Hygiene*, 9: 503, 1927.)
[13]Bohr, von Chr., and Maar, Vilh: Uber der Einfluss der Ozoneinathmung auf die Function der Lunge. (*Skand. Arch. f. Physiol.*, 16: 41, 1904.)
[14]Ewell, A. W.: Present Use and Future Prospects of Ozone in Food Storage. (*Food Research*, 3: 101, 1938.)
[15]Hartman, F. E.: The Industrial Application of Ozone. (ASHVE Transactions, 31: 33, 1925.)

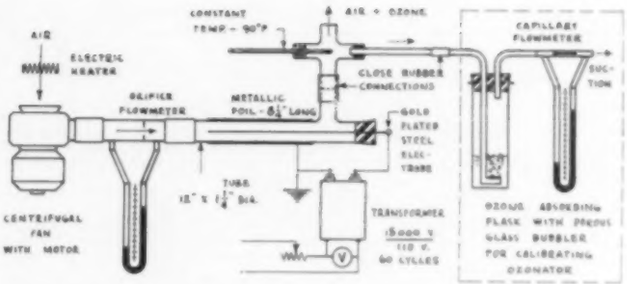

Fig. 1—Apparatus for producing and measuring ozone

kept closed in order to avoid inward leakage and the observations of odor intensity were made in the exhaust outlet outside the experimental room. With high airflow, door leakage was outward and did not affect the results.

Both rooms were ventilated by downward supply systems, and the quantity of air introduced was measured by thin-plate orifice meters. The used air escaped to the corridor through sensitive check louvers near the floor of the rooms. Air supply to the experimental room was varied in different experiments, according to the number of occupants and the desired body-odor intensity. In the control room, the air supply was always high enough to keep the body-odor intensity at or below the olfactory threshold. Separate air conditioners installed in the corridor kept the two rooms at a comfortable temperature and humidity.

Ozone Production and Measurement: Ozone was generated inside the experimental room by corona discharge using an electric precipitator similar to that of Drinker's for the quantitative determination of atmospheric dusts.[16] The air to be ozonized was filtered and heated always to about 90 F, as shown in Fig. 1. All parts of the apparatus were kept scrupulously clean to preclude production of nitrogen oxides.

The output of the ozonator was measured by oxidation of potassium iodide and titration of the liberated iodine with sodium thiosulfate using starch as an indicator.[17] The arrangement of apparatus for producing and measuring the ozone is shown diagrammatically in Fig. 1. Ozone output was controlled by varying the transformer primary voltage, the airflow through the ozonator, and the length of the inner electrode. Fifty-five different sets of conditions yielded the following empirical equation from which the ozonator output could be predicted within the experimental range:

$$\text{Cfm } O_3 = 53\,(V - 75)\,(250 - G)\,(Q + 25)\,(L + 8) \times 10^{-13}$$

where:

V = transformer primary voltage (80 to 110)
G = absolute humidity, grains per pound dry air (4 to 120)
Q = airflow through ozonator, cubic feet per minute (2 to 10)
L = length of precipitator inner electrode, inches (4 to 8)

The ozonator was located in the back of the experi-

[16]Drinker, Philip, et al: Alternating Current Precipitators for Sanitary Air Analysis. (*Journal Industrial Hygiene*, 14: 364, 1932.)
[17]Treadwell and Hall: Analytical Chemistry. (Vol. 2, Quantitative, 6th ed., John Wiley & Sons, New York, 1924.)

Table 2—Olfactory Scale of Odor Intensities

Odor Intensity Index	Designation	Explanation or Qualification
0	None	No perceptible odor
½	Threshold for keen sense of smell	Very faint, barely detectable by trained persons, imperceptible to untrained persons
1	Normal threshold	Detectable by all normal persons but not objectionable
2	Moderate	Neither pleasant, nor disagreeable; allowable limit in rooms
3	Strong	Objectionable; air regarded with disfavor
4	Very strong	Disagreeable, annoying, irritating
5	Overpowering	Sickening, nauseating

mental room behind the subjects, and the ozonized air was blown upward against the incoming air so as to obtain good mixing.

Results

Olfactory Ozone Intensity in Relation to Ozone Concentration: The success of the whole study depended largely on the ability of the human nose to judge low ozone concentrations that are below the limits of present chemical methods of analysis. It was necessary, therefore, to study first of all the relationship between olfactory intensity and concentration of ozone in clean air free from body odor.

Ozonized air in measured quantities and concentrations was diluted to known extents with the air supplied to the room, the mixing taking place inside the experimental room. After allowing sufficient time for equilibrium, two trained *judges* passed from the control to the experimental room several times in each test and recorded the strength of ozone, immediately after entering, according to the scale in Table 2. The results are shown in Fig. 2, and it will be seen that for any given ozonator output, the ozone concentration, and hence its olfactory intensity in the room, depended largely on the absolute humidity which affects the ozone decomposition rate. In a series of tests with absolute humidities of 3 to 5 grains per pound of air, obtained by silica-gel dehydration, ozone was found to be fairly stable after equilibrium was reached and the concentration in the air was but slightly lower than that computed from the ozonator output. This agrees with previous work on ozone decomposition in *bone-dry* air free from oxidizable material.[18]

Approximate ozone concentrations corresponding to any given ozone intensity in the experimental room can, therefore, be read off the 0 humidity line in Fig. 2. The amount of ozone to be added to the air in the room in order to obtain the given ozone intensity, or concentration, depends on the absolute humidity and can be read off the appropriate humidity line in Fig. 2. For example, to obtain an odor intensity of 2, the required ozone concentration must be about 0.03 ppm. If the humidity is 40 grains per pound of dry air the amount of ozone to be added to the air to obtain an equilibrium concentration of 0.03 ppm with a corresponding olfactory intensity of 2, is 0.10 ppm, whereas with a humidity of 100 grains per pound the ozone to be added must be approximately 0.9 ppm. These data apply only to the

[18]Schumacher, Hans-Joachim, and Sprenger, G.: Der Thermische Ozonzerfall. (*Zeitschr. f. Physiol. Chemie*, Abt. B, 6: 446, 1929-1930.)

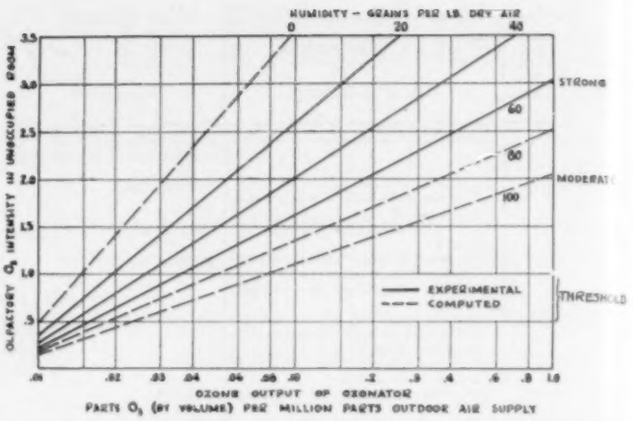

Fig. 2—Effect of humidity on disappearance of ozone in unoccupied experimental room

The lines for 0, 80, and 100 grains humidity were extrapolated from the data for the 4, 20, 40, and 60-grain lines using the following empirical equation:

$$R = 3.08 - \frac{\log (13.3\,G + 250)}{1.155}$$

in which G represents the humidity in grains per pound of dry air and R represents the ratio between ozone intensity at humidity G and ozone intensity at zero humidity on the same ordinate. The average deviation of the experimental data from the values given by this equation was 0.07 on the odor scale, and the range of variations was from 0.12 positive to 0.23 negative on that scale.

experimental room used in this study under conditions of equilibrium.

Neutralization of Body Odors by Ozone: The method of carrying out tests was somewhat similar to that followed in previous work on ventilation requirements[2, 3], except for the addition of ozone near the middle of each test. Experiments ran from 9 a. m. to 12:30 p. m. or 1 to 5 p. m. Altogether, 95 persons served as subjects, largely adults and grade-school children chosen from the poorest socio-economic status.

After the body-odor intensity was raised to equilibrium during the first part of each test, the ozonator was turned on and adjusted so as to get maximum body-odor obliteration with the minimum smell of ozone. Ozone and body-odor intensities were estimated by two to six *judges* upon entering the experimental room from the control room, using the olfactory scale in Table 2 for both odors.

Fig. 3 shows the progress of a typical experiment. The ozonator was turned on about an hour after the subjects entered, and in 20 min the body-odor intensity dropped from 3.5 to 0.5, while the ozone intensity kept on rising toward the objectionable point. When the ozone output was reduced, the body-odor intensity rose simultaneously, and after about an hour, both approached equilibrium.

A summary of the data obtained in fourteen tests under equilibrium conditions is given in Table 3. It can be seen in columns 6 and 7 that the introduction of ozone reduced the body-odor intensity in all instances, but in no case was it possible to counteract the two odors so that neither could be smelled. Ozone concentration had to be perceptible to do any good at all, and the higher the body-odor intensity, the higher the ozone intensity had to be to counteract it. The limiting factor was the ozone odor itself which sometimes became as objectionable as the body odor.

Table 3—Counteraction of Body Odors with Ozone

SUMMARY OF EXPERIMENTAL DATA UNDER CONDITIONS OF EQUILIBRIUM

1 Expt. No.	2 No. of Subjects	3 Air Temp. Deg F	4 Humidity Grains H_2O/lb Dry Air	5 Outdoor Air Supply cfm/ Person	6 Initial Body Odor Intensity (No O_3)	7 Final Body Odor Intensity (with O_3)	8 Observed Olfactory Ozone Intensity	9 Ozonator Output ppm. by Vol. Outdoor Air Supply	10 Estimated Ozone Concentration in Room Air ppm. by Vol. (from Fig. 2)
82	7	78.0	33	17	2.0	0	1.75	0.044	0.023
83	7	75.4	30	13	3.0	0	2.25	0.087	0.036
84	7	82.0	77	21	1.5	1.5	0.50	0.028	0.012
85	7	85.0	98	7	3.0	2.25	0.75	0.039	0.012
86	9	82.7	70	10	3.0	1.75	0.75	0.026	0.012
87	7	85.5	87	13	3.25	2.0	1.00	0.024	0.011
88	7	78.8	48	6	3.5	2.0	1.00	0.054	0.020
89	7	81.5	65	6	3.5	0.75	1.50	0.083	0.020
90	8	83.5	90	15	3.5	1.5	0.75	0.074	0.016
91	15	81.0	78	< 5	4.25	0.5	2.50	0.412	0.034
92	7	82.0	88	6	4.0	1.5	1.75	0.183	0.021
93	7	81.2	58	15	3.25	3.5	0.50	0.021	0.012
94	7	81.5	80	16	2.5	2.5	0.50	0.028	0.012
95	7	77.0	74	22	2.5	1.0	0.88	0.016	0.010

It would, therefore, seem that the use of ozone alone is impractical for counteracting strong body odors and that it must be supplemented by the introduction of sufficient outside air to lower the body-odor intensity to 3.5 or less. Small quantities of ozone can then be used to advantage for reducing the odor from 3.5 to 2 or less. Under the experimental conditions, an ozone odor intensity of 0.75-1.00, (0.015 ppm approximately) reduced the outdoor air requirement for the control of body odor more than 50 per cent, as can be seen in Fig. 4. Ozone intensities less than 0.75-1.00 had no effect on body odors, while higher intensities proved to be unpleasant to a few sensitive individuals.

Masking vs. Oxidation: The results of this study would seem to indicate that the action of ozone on body odors takes place not in the air of the room by oxidation, as is sometimes assumed, but on the mucous membranes of the nasal passages where the ozone is decomposed. If the reduction were due to oxidation in the air, some of the ozone would have been utilized in the process and the ozone intensity would have been lower with subjects than without subjects in the room, when all other conditions remained unaltered. No significant loss of ozone appears to have taken place, as can be seen in columns 5, 6, and 7 of Table 4. Positive signs in column 7 indicate loss and negative signs indicate gains of ozone, the latter due, of course, to errors. The mean loss in all 14 experiments is practically zero, in fact negative, indicating no detectable chemical action.

If the odorous emanations of the human body consist largely of acids, as is generally believed, little oxidation can be expected from the small quantities of ozone used in the experiments. On the other hand, the olfactory threshold concentration of ozone is low enough, by comparison with that of most body odors, to readily mask the latter, with the possible exception of the odors of skatol and indol.

Even hydrogen sulfide, which appears to be the most readily oxidizable substance in the known group of body odors, can only be masked in ventilation work because the concentration of ozone required for complete oxidation is prohibitive from the standpoint of both toxicity and smell. The normal olfactory threshold (odor intensity 1) of hydrogen sulfide is placed at 0.002 mg per liter of air. An objectionable odor strength of 3 would, therefore, correspond to $0.002 \times 2^3 = 0.008$* mg per liter approximately. To oxidize this hydrogen sulfide would theoretically require

$$0.008 \times \frac{48\dagger}{34\ddagger} = 0.0113 \text{ mg}$$

of ozone per liter of air, or $0.0113 \times 509 = 5.8$ ppm. Actually the amount of ozone required for complete oxidation must be considerably in excess of this already prohibitive concentration. On the other hand, about 0.1 ppm of ozone (0.0002 mg per liter) appears to be sufficient to mask a hydrogen sulfide odor intensity of 3. This quantity of ozone could theoretically oxidize a maximum of $\frac{34}{48} \times 0.0002 = 0.00014$ mg of hydrogen sulfide or only $\frac{0.00014}{0.0113} \times 100 = 12$ per cent of the hydrogen sulfide in the air.

Using even much greater ozone concentration in an experimental chamber, Erlandsen and Schwarz[4] failed to obtain any demonstrable oxidation of hydrogen sulfide, ammonia, trimethylamine, butyric and valeric acids, indol and skatol, although the odors of these substances were effectively masked.

Further evidence in support of the *masking* effect is the reappearance of body odors with disappearing ozone odor at the end of an experiment (see Fig. 5) after the subjects had left the room—the ozonator and air supply having been turned off and the room closed tight. Referring to Fig. 5, the ozone intensity just before the subjects left was high enough to obliterate completely the body odor, but when the ozone intensity fell to the threshold, the body odor reappeared and continued to increase, gradually approaching the normal body-odor disappearance curve (dotted line) without ozone. The implication here is that ozone was incapable of destroying

*Concentration corresponding to an odor intensity index n = olfactory threshold concentration $\times 2^{n-1}$.

†48 = molecular weight of ozone.

‡34 = molecular weight of hydrogen sulfide.

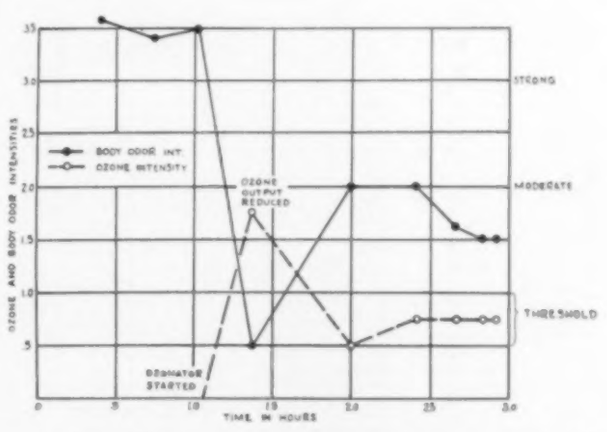

Fig. 3—Record of an experiment showing effect of ozone odor on body-odor intensity. Room air supply and number of occupants constant

body odors in the air while the subjects were in the room, despite its high concentration. Odoriferous material was adsorbed on wall surfaces and when the subjects left, desorption began and continued at its usual normal rate, as if no ozone had been added at all.

Possibilities and Limitations

From the standpoint of ventilation, it should make no great difference whether ozone masks or actually destroys body odors in the air. If it improves the quality of air and does no harm to the occupants, it should be of some value, and under favorable conditions it has proved to be of value in this and other previous studies.[19]

Although the *judges* were extremely sensitive to the odor of ozone upon entering the experimental room (containing ozone) from the control room (containing no ozone), most of the subjects in the experimental room were unaware of ozone intensities of 1.0 or less corresponding to ozone concentrations of 0.015 ppm or less, except when the concentration was built up too rapidly. Many subjects noticed an improvement in air freshness at such low ozone concentrations, while a few complained of a *strange odor in the air*.

Ozone intensities over 2 (0.04 ppm or more) were associated with dryness of the upper respiratory mucosa, and distinct irritation at higher concentrations. It appears, however, from Table 3 and Fig. 4, that it is rarely necessary to use ozone intensities greater than 1.0 (0.015 ppm). This concentration can barely be smelled by the occupants of a room, yet it permits a considerable reduction in the fresh-air supply (see Fig. 4) and there is no evidence that this amount is harmful even on prolonged exposures, as for instance in the London subways.

The main objection to the application of ozone to ventilation is lack of control. Although the output of commercial ozonators may be controlled adequately by dehydrating and filtering the treated air, no means are available for regulating the actual room concentration which may vary within wide limits according to humidity (see Fig. 2) and general characteristics of a room. A given ozone output at the machine may yield only a trace of ozone in a room on a humid day, but the same output in dry weather may result in a most undesirable condition. The danger is especially great in central station systems that have sufficient capacity to make up for loss of ozone in transit to the rooms.

[19]Hallett, E. S.: Ozone in the St. Louis Schools. (ASHVE Transactions, 28: 33, 1922.)

Table 4—Loss of Ozone-Odor Intensity in Counteracting Body Odors

1 Expt. No.	2 Room Absol. Humidity Grains/lb Air	3 Ozonator Output ppm. by Vol. Outside Air Supply	4 Estimated Ozone Concentration in Room Air ppm. by Vol. (from Fig. 2)	5 Observed Ozone Intensity with Subjects in Room (from Col. 8, Table 3)	6 Corresponding Ozone Intensity with no Subjects in Room (from Fig. 2)	7 Loss of Ozone Intensity (Col. 6 − Col. 5)
82	33	0.044	0.023	1.75	1.60	−0.15
83	30	0.087	0.036	2.25	2.20	−0.05
84	77	0.028	0.012	0.50	0.70	+0.20
85	98	0.039	0.012	0.75	0.70	−0.05
86	70	0.026	0.012	0.75	0.70	−0.05
87	87	0.024	0.011	1.00	0.60	−0.40
88	48	0.054	0.020	1.00	1.40	+0.40
89	65	0.083	0.020	1.50	1.40	−0.10
90	90	0.074	0.016	0.75	1.10	+0.35
91	78	0.412	0.034	2.50	2.10	−0.40
92	88	0.183	0.021	1.75	1.50	−0.25
93	58	0.021	0.012	0.50	0.70	+0.20
94	80	0.028	0.012	0.50	0.70	+0.20
95	74	0.016	0.010	0.88	0.40	−0.48
Mean				1.17	1.13	−0.04

Under such conditions, ozone may do more harm than good and its use should be discouraged, at least until adequate means are devised for accurately controlling the concentration in occupied rooms. The so-called *ionizing* machines which have recently appeared produce less ozone than the old ozonators and their effectiveness for *masking* odors is correspondingly smaller. The ions produced by these machines have little effect on odors and *freshness* of air. The underlying agent is ozone and the problem of concentration control still remains unsolved.

Summary

Ozone in concentrations of 0.015 ppm, that can barely be smelled by the occupants of a room, reduced the smell of body odor sufficiently to permit a reduction of at least 50 per cent in the fresh-air requirement for odor control. Higher concentrations were irritating to the mucous membranes of the upper respiratory tract, while lower concentrations had no effect on body odors. The action of ozone on body odors appears to take place not in the air of the room by oxidation, as is sometimes assumed, but on the mucous membranes of the nasal passages by

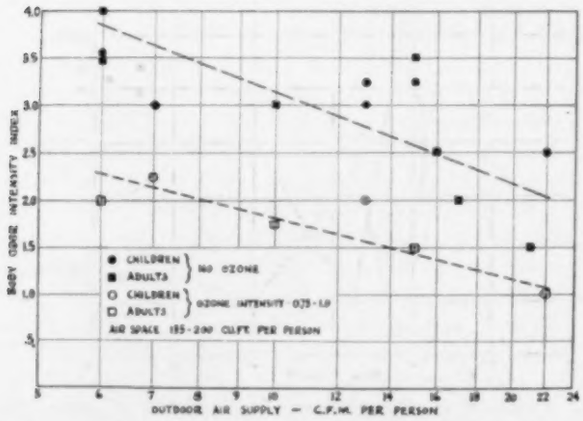

Fig. 4—Effect of a mild ozone odor on body-odor intensity and on air requirements for body-odor control

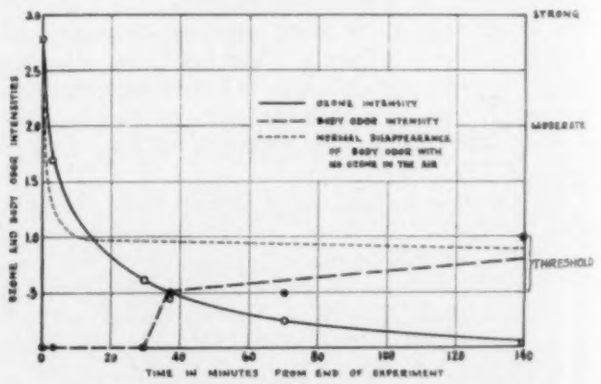

Fig. 5—Reappearance of body odor with disappearing ozone odor in tightly-shut room at the end of an experiment after subjects left room

masking. Although ozone in sufficient strength completely obliterated the smell of body odors, it proved difficult to counteract the two so that neither could be smelled by persons entering the room from fresh air. Ozone had to be present in perceptible concentrations to do any good at all.

The main objection to the application of ozone to ventilation is shown to be lack of control. Until adequate means are devised for controlling the output of ozonators from the actual concentration in occupied spaces, the use of ozone should be discouraged because of its great toxicity.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the ASHVE Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, ASHVE Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

Portfolio of Air Conditioning in Hospitals. *The Modern Hospital,* Vol. 53, No. 1, July 1939, pp. 49-64. Group of five papers: (1) Advantages and Limitations of Hospital Air Conditioning, by C. P. Yaglou; (2) Physiological Aspects of Air Conditioning, by M. B. Ferderber, A. A. Rosenberg, and F. C. Houghten; (3) Air Conditioning in a 52-Bed Hospital, by R. D. Brisbane; (4) A Trustee Considers Air Conditioning, by Charles F. Neergaard; (5) Allergy Room Is Appreciated, by Lucius R. Wilson.

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Effect of Storage Conditions Upon Dust-allaying Coal Treatment, by F. W. Godwin. *Coal-Heat,* Vol. 36, No. 2, August 1939, pp. 16-18. Description of dust cabinet and method of its use in determining dustiness of coal samples. Data on dust index after cabinet treatment storage at various relative humidities and after coal aid treatment.

•

Correcting Excess Air Leakage, by Julius Wolf. *Coal-Heat,* Vol. 36, No. 2, August 1939, pp. 34-36. Discussion of ideal air supply for stoker operated furnace and harmful results of excess air or air leakage through dead plates.

•

Measurement of the Fineness of Powdered Materials, by Harold Heywood. *The Institution of Mechanical Engineers Proceedings,* Vol. 140, 1938, pp. 257-347. (London.) Description in detail of methods of analysis available for measuring the fineness of powdered materials used in engineering and industrial processes. Definitions of particle size and shape are discussed. Size grading by sieving, elutriation by means of air or water sedimentation, hydrometer readings and the obscuring of a beam of light by dilute suspension of the particles are discussed in detail.

•

Modern Methods of Marine Heating and Ventilation, by David M. French. *The Heating and Ventilating Engineer,* Vol. XII, No. 144, June 1939, pp. 550-554, (London). Discussion of some factors affecting heating and ventilating systems on board passenger ships. Advantages of steam and hot water heating in different types of ships. Special problems of heating and ventilating in ships discussed. Air conditioning considered essential in passenger ships crossing the equator.

•

The Globe Thermometer, by J. H. Brunklaus. *The Heating and Ventilating Engineer,* Vol. XIII, No. 146, August 1939, pp. 61-65 and 85, (London). Mathematical discussion of accurate measurement of air temperature. Description of globe thermometer and its use to measure the degree of comfort in different points of a room at the same time. Discussion of heat loss by the human body by evaporation, convection and radiation.

•

British Standard Code for the Testing of Fans for General Use; issued as B. S. No. 848—1939. This code may be had from the British Standards Institute, 28 Victoria Street, London, S.W.1; 2s each, or 2s 2d post free.

•

Solving Draft Problems. Skillful Application of Draft Gauge Reveals Opportunities to Cut Oil Bills, by W. B. Adams. *Air Conditioning and Oil Heat,* Vol. XII, No. 8, August 1939, pp. 10 and 11, and 28-30. Discussion of use of the draft gauge to improve operating conditions in an oil furnace. Normal readings of draft test data for coal-fired boiler and for oil-fired boiler.

•

Summer Conditioning Hours, a Comparative Measure of Summer Air Conditioning Requirements. *Air Conditioning and Oil Heat,* Vol. XII, No. 8, pp. 15-16, August 1939. Summer conditioning hours are the number of hours during which natural atmospheric conditions are such as to cause discomfort according to the Comfort Chart developed by the ASHVE, the U. S. Bureau of Mines, and the U. S. Public Health Service. Summer conditioning hours and summer conditioning days have been computed for April, May, and June, 1939.

•

Ice in Air Conditioning, by O. W. Kothe. *Automatic Heat and Air Conditioning,* Vol. 11, No. 2, August 1939, pp. 16-18 and 41-43. Construction and performance features of ice-air conditioners. Conditions to be expected from this type of unit and plan drawings to simplify the more technical aspects of the discussion.

•

Air Conditioned Mushrooms. *Automatic Heat and Air Conditioning,* Vol. 11, No. 2, August, 1939, p. 18. Description of a typical, large air conditioning installation for mushroom houses. The equipment is illustrated.

•

Night Air Is Cool Air. *Plumbing and Heating Journal,* Vol. 107, No. 2, pp. 20-21. Description of attic ventilation fans and the use of such fans in air conditioning houses or stores.

•

How to Install an Attic Fan, by T. A. Wallace. *Coal-Heat,* Vol. 36, No. 1, July 1939, pp. 46-48. Discussion of some elementary principles in cooling by ventilation, with suggestions regarding analysis and calculations for typical applications and directions for installation of various types of attic ventilation fans.

Atlanta Skyline from Terminal Station Plaza

Society Goes to Atlanta

ATLANTA Chapter members are awaiting the Fall Meeting of the Society and have prepared an interesting program for those who attend this meeting scheduled for October 30 and 31 at the Atlanta Biltmore Hotel, Atlanta, Ga.

There are to be three technical sessions with nine papers: two papers give results of investigations carried on under the cooperative research program sponsored by the Society's Committee on Research; two papers specifically relate to problems in the South; two more are concerned with industrial research subjects; and one is devoted to business relationships. In addition to these technical papers, a program of social events is scheduled for each day.

After registration on Monday morning, October 30, there will be a general get-together luncheon for members and ladies in the Atlanta Biltmore at 12:15 p. m. and the guest of honor will be Dr. M. L. Brittain, president of Georgia School of Technology. During the afternoon a sightseeing trip for ladies has been planned and at the conclusion of the afternoon technical session there will be a social hour for members and ladies on the terrace of the Atlanta Biltmore. A dinner and dance will be the feature of the evening and the guest of honor will be Robert T. Jones, Atlanta attorney, who is more generally known as Bobby Jones, the golfer. The speaker will be O. B. Keeler, Atlanta sports commentator.

On Tuesday the ladies will have a bridge party during the morning and the afternoon will be devoted to various inspection trips.

The location of Atlanta, slightly west of Cincinnati and Detroit, is within easy reach of a large percentage of the Society membership. There is excellent train service from all major cities and fine roads lead to Atlanta for those who desire to motor. En route to Atlanta there is much of scenic beauty, particularly for those coming from the North and West.

The distance from Atlanta to some of the principal cities is indicated in the following table:

Birmingham, Ala.	168	miles
Charlotte, N. C.	260	"
Nashville, Tenn.	288	"
Greensboro, N. C.	353	"
Mobile, Ala.	354	"
Louisville, Ky.	453	"
Cincinnati, O.	488	"
New Orleans, La.	495	"
St. Louis, Mo.	611	"
Washington, D. C.	637	"
Miami, Fla.	705	"
Cleveland, O.	749	"
Detroit, Mich.	757	"
Philadelphia, Pa.	778	"
Chicago, Ill.	789	"
Pittsburgh, Pa.	796	"
Dallas, Tex.	826	"
Houston, Tex.	857	"
New York, N. Y.	863	"
Kansas City, Mo.	905	"
Boston, Mass.	1093	"

Reduced round trip rates are offered by railroads and through-car service and fast schedules are available to Atlanta from various Chapter cities.

The headquarters hotel offers special rates on the European plan: single room and bath—$3.00; double room for two persons—$4.50; double room, twin beds—$5.00 per day.

Society members should tune in on the special radio broadcast on October 26 at 10:30 p.m. standard time, Station WSV, as special features of the meeting will be broadcast on the Welcome South, Brother Program.

The Committee in charge of arrangements is looking forward to a large turnout of members and has everything in readiness for the Atlanta meeting. For early arrivals there will be an opportunity to see a football game at the Georgia

C. L. Templin
Reception

T. T. Tucker
Ladies

L. F. Kent
General Chairman

C. T. Baker
Vice-Chairman

E. W. Klein
Banquet

Tech Stadium, Saturday, October 28, and there will be golf for those who desire to play. Prior to the opening of the meeting there will be a meeting of the Council of the Society.

Members of the Atlanta Chapter and their wives will serve as hosts to the Society and events will be in charge of the Committee on Arrangements, of which L. F. Kent is General Chairman.

PROGRAM—Fall Meeting

American Society of Heating and Ventilating Engineers
Atlanta Biltmore Hotel, Atlanta, Ga. October 30-31, 1939

October 28

1:00 p.m. Reception of early arrivals
2:00 p.m. Football Game—Georgia Tech Stadium

October 29

10:00 a.m. Informal Golf Matches
11:00 a.m. Council Meeting

October 30

8:30 a.m. Registration
9:30 a.m. Technical Session—
The Use of Air Velocity Meters, by Prof. G. L. Tuve
Temperature Droop Characteristics of Room Thermostats, by B. E. Shaw
12:15 p.m. Get-together Luncheon (Members and Ladies)
Speaker—Dr. M. L. Brittain, President Georgia Tech
1:45 p.m. Ladies Sightseeing Trip
2:00 p.m. Technical Session—
Food Preservation in the South, by C. T. Baker
Application of Storage Refrigeration to Air Conditioning, by C. F. Boester
A Standard Air Filter Test Dust, by Prof. F. B. Rowley
An Analysis of the Air Conditioning Contractor's Business Relationships, by John H. Carter
4:30 p.m. Friendship Hour for Members and Ladies (Biltmore Terrace)
7:00 p.m. Autumn Dinner and Dance
Guest of Honor—Robert T. Jones—Atlanta Attorney and Golfer extraordinary
Speaker—O. B. Keeler—Sports Commentator and Member of Collier's All American Board

October 31

9:30 a.m. Technical Session—
Heating and Ventilating Problems as Related to Agriculture, by Prof. F. B. Lanham
A Laboratory for Use in Determining the Influence of Various Temperatures and Relative Humidities on the Final Quality of Air-Cured Tobacco, by Prof. L. S. O'Bannon
Maximum Permissible Humidities in Humidified Buildings, by Paul D. Close
10:00 a.m. Ladies Breakfast-Bridge
1:30 p.m. Inspection Trips
2:00 p.m. Golf Tournament—East Lake Country Club

Stone Mountain

O. B. Keeler
Banquet Speaker

Bobby Jones
Guest of Honor

Cups (*left to right*)—British Amateur, British Open, Walker Cup Team Trophy, U. S. Open, U. S. Amateur

Committee on Arrangements

General Chairman—Laurence F. Kent

Vice-Chairman—C. T. Baker

Banquet, E. W. Klein, *Chairman*, W. P. Sherman, Leo Sudderth, Jr., J. W. Parker.

Finance, C. B. Cole, *Chairman*, R. F. Hahn, L. F. Lawrence, Jr., R. E. Nolan, C. S. Minor, Jr., Lewis A. Hamilton.

Ladies, T. T. Tucker, *Chairman*, E. R. Foss, Spotswood Parker.

Publicity, M. M. Crout, *Chairman*.

Reception, C. L. Templin, *Chairman*, W. J. McKinney, F. L. Laseter, Blackman Dunn, R. E. Vernon, Carl Johnson.

Registration, S. W. Boyd, *Chairman*, L. L. Barnes, M. G. Driscoll, R. B. Stotz.

Sports, A. H. Koch, *Chairman*, G. H. Brodnax, Jr., G. T. Gunnell, C. L. Schroeder, Jr.

Transportation & Tours, C. E. Brockinton, *Chairman*, L. L. Hamilton, R. D. Kelley, T. R. Yarborough, L. H. Hobson.

Naval and Society Engineers Meet at Bremerton

A meeting of engineers from the Navy Department, including Naval Officers, as well as members of the Seattle Chapter of the Society, was held on August 28, 1939, at Bremerton, Wash., with 60 in attendance. Dinner was served preceding the meeting which was arranged by Foster J. Pratt, chairman of arrangements and member of the Seattle Chapter. Mr. Pratt introduced the speakers of the evening, William Goodman, of LaCrosse, Wis., and Herbert Kunen, of New York City. Mr. Goodman gave an interesting presentation on heat transfer coils and psychrometric charts, while Mr. Kunen discussed air flow in ducts and distribution, both of which brought about a lively and instructive discussion.

New Executive Committee Appointments

Pres. J. F. McIntire, announces the appointment of Dr. F. E. Giesecke as Chairman of the Executive Committee due to the resignation of E. Holt Gurney, Toronto because of his war service responsibilities. He has also announced the appointment of Prof. E. O. Eastwood, Seattle, Wash. as a member to serve for the remainder of this year. The Committee personnel is F. E. Giesecke, *Chairman*, W. L. Fleisher, and E. O. Eastwood.

NOMINATIONS FOR 1940

Nominations for Officers and Council

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1940, submits the following list of nominees:

For President:
F. E. GIESECKE, College Station, Texas

For First Vice-President:
W. L. FLEISHER, New York, N. Y.

For Second Vice-President:
E. O. EASTWOOD, Seattle, Wash.

For Treasurer:
M. F. BLANKIN, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

J. F. S. COLLINS, JR., Pittsburgh, Pa.
E. N. MCDONNELL, Chicago, Ill.
T. H. URDAHL, Washington, D. C.
C.-E. A. WINSLOW, New Haven, Conn.

Respectfully submitted,
NOMINATING COMMITTEE,
H. H. ERICKSON, *Chairman*
TOM BROWN, *Acting Secretary.*

In accordance with the provisions of the Society's Constitution, By-Laws and Rules, ballots containing the names of the above candidates will be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. B-VIII—Section 11. The Nominating Committee shall consist of one (1) member eligible to vote designated by each Chapter, or his alternate also appointed by the Chapter. The Secretary of each Chapter shall certify to the Secretary of the Society on or before January first the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September twentieth, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the JOURNAL.

Art. B-IX—Section 2. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, eligible to vote, at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election.* The election is governed by the By-Laws for the election of officers, with the single exception that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the Government of the Research Laboratory, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1940:

Three-Year Term

PHILIP DRINKER, Boston, Mass.
AXEL MARIN, Ann Arbor, Mich.
A. E. STACEY, JR., New York, N. Y.
J. H. VAN ALSBURG, Chicago, Ill.
J. H. WALKER, Detroit, Mich.

The regulations governing the nomination and election of members of the Committee on Research are as follows

ARTICLE II—ORGANIZATION

Section 1. *Committee on Research*—There shall be a standing committee known as the Committee on Research, consisting of fifteen (15) members each serving for three (3) years and five (5) retiring each year. The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall without election become an additional member of the Committee on Research for one (1) year.

(*a*) The Council shall nominate previous to July first of each year five (5) members to fill the vacancies of those retiring at the next Annual Meeting.

(*b*) The nominations made by the Council shall be published in the October issue of the Society's JOURNAL.

(*c*) Any ten (10) members of the Society eligible to vote may present to the Secretary over their signatures, the name of one (1) or more additional nominees for the Committee on Research, provided such name or names are presented at least sixty (60) days prior to the next Annual Meeting, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

(*d*) The election shall otherwise conform to the regulations provided for the election of officers of the Society in the Constitution, By-Laws and Rules.

(*e*) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Arrangements Committee for 46th Annual Meeting Announced

The Cleveland Chapter will be host to the Society for its 46th Annual Meeting to be held at the Hotel Statler during the week of January 22, and announcement of the personnel of the Committee on Arrangements has just been made as follows: L. T. Avery, General Chairman, G. L. Tuve, Honorary Chairman, Philip Cohen, Vice Chairman and L. S. Ries, Vice Chairman; *Technical Sessions*, C. F. Eveleth, chairman; *Publicity*, John Paul Jones, chairman; *Finance*, E. W. Gray, chairman; *Inspection Trips*, C. A. McKeeman, chairman; *Attendance*, D. L. Taze, chairman; *Banquet*, F. A. Kitchen, chairman; *Transportation*, A. L. Vanderhoof, chairman; *Entertainment*, H. E. Wetzell, chairman; *Registration-Reception*, Paul Gayman, chairman; and *Ladies*, Mrs. W. R. Rhoton, chairman.

L. T. Avery
General Chairman

During the fourth week of January 1940 the *National Warm Air Heating and Air Conditioning Association* will hold its mid-year meeting, and the 6th International Heating and Ventilating Exposition will be housed in Lakeside Hall.

The indicated extent and comprehensiveness of the displays point to the largest heating and ventilating exposition ever held. The exhibitors will present the latest methods and apparatus developed in the field of heating, ventilating and air conditioning and in refrigeration equipment related to air conditioning. Nearly 300 manufacturers have already engaged exhibit space, using more than 75 per cent of the available space in Lakeside Hall.

The exposition will bring together exhibitors showing the best that the 20th century has developed in this field. There will be many new types of instruments for control and regulation and a host of auxiliary products of interest to everyone. The exposition will be under the personal direction of C. F. Roth, president of the International Exposition Co., who has been responsible for each of the five previous expositions.

E. R. Downe Joins Bryant

The appointment of E. R. Downe as chief engineer of The Bryant Heater Co. has just been announced by F. R. Higley, director of research and development. Mr. Downe was formerly vice-president and chief engineer of the American Gas Products Division of the American Radiator Co., New York, N. Y., and has a background of 29 years of experience in engineering and designing of heating equipment. He will make his headquarters in Cleveland. Mr. Downe is a member of the ASHVE, *AGA* and *ASME*.

Speakers Bureau Active

With the opening of the meeting season for local chapters the Speakers Bureau, established by the Council, has received many requests for speakers. The Committee work is directed by E. N. McDonnell, Chicago, chairman, and he is assisted by John Howatt, Chicago, W. G. Boales, Detroit, and W. A. Russell, Kansas City, Mo. Members who are scheduled to appear under the auspices of the Speakers Bureau are: A. E. Stacey, Jr., at the Michigan Chapter, October 16; Philip Drinker, at the Philadelphia Chapter meeting November 9; and S. R. Lewis, before the Cleveland meeting November 13.

Society Officers Will Speak at Chapter Meetings

Speaking engagements will be filled by Society Officers during the fall season and W. L. Fleisher, second Vice-President and Chairman of the Committee on Research will address a meeting of the St. Louis Chapter of the Society and the Engineers Club of St. Louis on October 5. On November 13 he will address the Pittsburgh Chapter.

Dr. F. E. Giesecke, first Vice-President of the Society, will visit eastern and middle western chapters during the months of November and December. His itinerary will include North Carolina on November 2, Washington, D. C., November 6, New York, November 20, Boston, November 21, Montreal, November 27, Toronto, November 28, Western Michigan, December 4, and Kansas City, December 11.

On September 1 Dr. Giesecke retired as Director of the Experimental Station at the Agricultural and Mechanical College of Texas, but will continue to serve as Professor of heating, ventilating and air conditioning. He will have charge of the summer short course in air conditioning, and also carry on research in his special fields of activity.

Eastern Air Conditioning Conference in November

On November 10 and 11 Lehigh University, Bethlehem, Pa., will sponsor an Eastern Air Conditioning Conference in cooperation with the *ASRE*, the ASHVE Philadelphia Chapter, and the *ACMA*. The Conference Committee in charge of arrangements is as follows: Prof. B. H. Jennings, *chairman;* W. A. Bornemann, W. H. Carrier, D. L. Fiske, W. B. Henderson, J. R. Hertzler, F. V. Larkin, S. R. Lewis, H. M. Roberts, J. L. Rosenmiller, M. C. Stuart, and C. C. Williams.

The tentative program is announced as follows:

November 10
Friday

9:00 a.m.—Registration at Packard Laboratory.

10:30 a.m.—*First Session*—Operation and Service for Air Conditioning Systems (Packard Auditorium). *Chairman*—A. R. Stevenson, Jr., Schenectady, N. Y.
Refrigeration-Storage for Handling Peak Loads by C. F. Boester.
Address of Welcome—C. C. Williams, President, Lehigh University.
Hotel Guestroom Air Conditioning Practices, by J. R. Hertzler.
An Analysis of Air Conditioning for an Educational Building, by S. R. Lewis.

12:45 p.m.—Luncheon on the Campus.

2:00 p.m.—*Second Session*: Human Responses to Comfort Air Conditioning (Packard Auditorium). *Chairman*—Prof. F. E. Giesecke, College Station, Texas.
Air as a Vehicle of Contagion, by Dr. W. F. Wells.
A Realistic Approach to Hospital Air Conditioning, by C. F. Neergard.
Uses of Air Conditioning in Hospitals, by M. B. Ferderber, M.D.

4:00-6:00 p.m.—Inspection of University Laboratories.

7:00 p.m.—Banquet and Get-together—Hotel Bethlehem, arranged by Phil:delphia Chapters of ASHVE and *ASRE*. Toastmaster—Crosby Field. Speaker—Philip Drinker.

November 11
Saturday

10:00 a.m.—*Third Session*: Industrial Air Conditioning (Packard Auditorium). *Chairman*—Dean A. C. Callen, Lehigh University.
Climate and Air Conditioning, by J. C. Albright.
Moisture Removal from Blast Furnace Air, by L. I. Lewis.
Creating a Demand for Air Conditioning, by J. L. Rosenmiller.
The Effect of Modified Atmospheres on Cold Storage, by Charles Brooks.

2:00 p.m.—Football—Lehigh vs. Muehlenberg, University Stadium.

ANNUAL REPORTS OF LOCAL CHAPTERS

Cincinnati

Cincinnati Chapter reports a successful year from all standpoints, with excellent speakers and well attended meetings. Social meetings and stag parties helped to achieve a notable advancement in the excellent spirit that prevailed throughout the year.

A resume of monthly meetings has been prepared by R. E. Kramig, Jr. and W. H. Junker.

The *October* meeting opened the season for Cincinnati Chapter. The speaker was J. F. McIntire, Detroit, president of the ASHVE, and his subject was The Engineer. Mr. McIntire's talk was an interesting one and the material was presented in a familiar manner.

The *November* meeting was devoted to a discussion of Codes. Mr. Stegner, of the Building Commission of Cincinnati, attended with his entire staff. The discussion centered around a new general heating code which was being developed. The outcome of the discussion which was started in this meeting was a new and comprehensive heating code for the City of Cincinnati.

The speaker at the *December* meeting was J. H. Walker, Detroit Edison Co. His discussion and explanation of the air conditioning system in a new glass brick building was of great interest to members.

The *January* meeting was a social gathering with a dinner dance at the Fox and the Crow. The only complaint heard after the evening was that the Chapter did not hold enough of the same type of meetings.

The *February* meeting was on Air Diffusion. F. J. Kurth, Anemostat Corp. of America, New York, was the speaker.

In *March* the meeting was held at the University of Cincinnati. Prof. G. L. Larson, University of Wisconsin, Madison, was the speaker on Mechanical Systems of Hot Water Heating.

The *April* meeting was turned over to Anker Winther, York Ice Machinery Corp., who spoke on the Application of Air Washers in Air Conditioning. In addition to an interesting talk Mr. Winther presented charts and diagrams. A moving picture, Cold Magic, followed the talk.

The *May* meeting closed the 1938-39 season and was held at the Homestead Country Club. Previous to the session the Chapter attended the baseball game between the Cincinnati Reds and the Boston Bees. After dinner a business meeting was held at which the election of officers took place.

The following were elected to serve for the 1939-40 season:

President—H. E. Sproull.

Vice-President—R. E. Kramig, Jr.

Secretary-Treasurer—W. H. Junker.

Board of Governors—H. E. Sproull, O. W. Motz, R. E. Kramig, Jr., W. C. Pistler and W. H. Junker.

Membership—R. E. Kramig, Jr., chairman; G. V. Sutfin, K. A. Wright and E. C. Williams.

Meetings—M. E. Mathewson, chairman; H. K. Jennings and H. A. Pillen.

Finance—W. C. Pistler, chairman; I. B. Helburn and E. B. Royer.

Legislative—O. W. Motz, chairman; C. E. Hust and A. L. Hard.

Southern California

Southern California Chapter opened the 1939-40 season with a meeting on September 12 when William Goodman of the Trane Co. was the speaker.

The Chapter has 53 members and held the following meetings during 1938-39.

October 12, 1938. Held at Colonial Banquet Room, Los Angeles, with 71 members and guests in attendance. Pres. H. M. Hendrickson appointed the following committees: *Membership*—O. W. Ott, Chairman; *Program*—J. F. Parke, Chairman; *Publicity*—H. H. Douglas, Chairman; and Speakers—E. H. Kendall, Chairman. Following the business meeting, Glen Miller, a heating and ventilating engineer, spoke on the small gas-fired refrigerating machines for air conditioning, which proved to be a worthwhile discussion.

November 9, 1938. The November meeting was also held in the Colonial Banquet Room in Los Angeles attended by 146 members and guests. President Hendrickson, after calling the meeting to order and presiding over the business session, introduced F. J. Kurth, who gave an interesting talk on air distribution. A lively discussion followed which was participated in most actively by those present.

December 16, 1938. The routine business having been taken care of, President Hendrickson introduced the speakers of the evening. The first presentation was given by Prof. L. M. K. Boelter, assisted by Prof. W. E. Mason, both of the University of California, who talked on the research work being performed on cooling towers, bringing before the meeting the latest available information. Professor Boelter was followed by Mr. Colesworthy, who gave a short talk and demonstration of the Westinghouse Precipitron filter. There was an attendance of 54 members and guests at this meeting.

January 3, 1939. The meeting was called to order by President Hendrickson, at the Colonial Banquet Room, with 62 members and guests attending. The regular order of business being attended to, W. L. Fleisher, Chairman of the Research Committee of the Society, was then introduced. Mr. Fleisher gave an interesting presentation of the problems that had to do with human comfort as applied to industry, which was followed by an open discussion.

February 28, 1939. The February meeting was held in Studio B in the National Broadcasting Studio, Hollywood. There being no business at this meeting, the President introduced J. F. Parke, Chairman of the Program Committee, who discussed the installation of the air conditioning system in this building, which was followed by a conducted tour through the Studios with special emphasis being placed on the air conditioning installation. There were 76 members and guests at this meeting.

March 23, 1939. This meeting was held at the Elks Club, Pasadena, with 86 members and guests present. A short business meeting was conducted, and on motion of Mr. Miller, a permanent Code Committee investigating heating, ventilating and air conditioning was established. Following the business session Prof. R. T. Knapp, California Institute of Technology, gave a talk on fan and blower characteristics.

April 11, 1939. The 51 members and guests who assembled for the April meeting returned to the Colonial Banquet Room, with President Hendrickson presiding. No business to be discussed, the meeting was turned over to the Program Committee, whose chairman introduced the guest speaker of the evening, J. F. McIntire, Vice-Pres. U. S. Radiator Corp., Detroit, and President of the Society. President McIntire delivered an interesting talk on The Birth of Air Conditioning, which was followed by an informal discussion concerning the affairs of the Society.

May 16, 1939. The May meeting was held at the Royal Palms Hotel with 61 members and guests in attendance. As customary the meeting was called to order by President Hendrickson, who called for a report from the Nominating Committee. This report having been sent to the Chapter by mail previous to the meeting, the President passed out ballots and conducted the election for the coming year and the following, approved by the Chapter, were elected to serve for the 1938-1939 season:

President—J. F. Parke.
Vice-President—H. H. Douglas.
Secretary—Earl Wells.
Treasurer—Glen Miller.
Board of Governors—H. M. Hendrickson, A. J. Hess, Walter Fabling and O. W. Ott.

Following the business meeting the speaker of the evening, Dr. B. F. Rayber, University of California, spoke on the evaluating air conditioning bids where no specifications were written up on the job on which the bids were based. This subject caused a most interesting discussion which lasted well on into the evening.

June 13, 1939. The June meeting of the Chapter was called to order by Pres. H. M. Hendrickson and the newly elected officers were installed with fitting ceremony. As there was no other business to come before the inspection trip was made through the plant of the Acme Brewing Co., the members were divided into small groups and guides explained the various mechanical devices and their operation.

After the inspection trip a Dutch lunch was enjoyed by the 77 members and guests present, after which the meeting was adjourned.

This meeting concluded the work of the Southern California Chapter of the Society for the year 1938-39.

Illinois Chapter

Illinois Chapter started the year with a membership of 159 and in spite of an unusually large number of resignations, due to transfer, finished the year with 158 members.

October 10, 1938. There were 81 members and guests at the opening meeting of the 1938-39 season. The guest speaker, J. R. Allen, chairman of the Research Committee on Industrial Hygiene for the *American Foundrymen's Association*, chose as his subject Ventilating Research in Industrial Hygiene.

November 14, 1938. At the November meeting S. R. Lewis, consulting engineer and Past President of the Society, gave an interesting talk on A Heating and Cooling Engineer Discusses Troubles. There were 120 members and guests in attendance.

December 12, 1938. The topic Health Aspects of Air Conditioning was ably given by Dr. R. W. Keeton, Head of Department of Medicine, University of Illinois, and Dr. F. K. Hick, also of the Department of Medicine. The meeting was attended by 84 members and guests.

January 9, 1939. Dr. Thomas C. Poulter, second-in-command of the Byrd Expedition to the South Pole and now director of research at Armour Institute, was guest speaker of the evening. He gave an interesting talk on the scientific work of the Byrd Expedition. There were 110 members and guests to hear Dr. Poulter.

February 13, 1939. The Four D's of Air Conditioning, ducts, dampers, distribution and deflection, was presented by J. H. Van Alsburg, of the Society's Speakers Bureau. There were 88 members and guests in attendance.

March 13, 1939. There were 92 at the March meeting, at which Air Conditioning Markets and Their Trends was discussed by K. C. Porter and William Rock, of the Commonwealth Edison Co.

April 10, 1939. An open and informal discussion took place on the new Chicago Building Code with special reference to those parts of the code pertaining to heating, ventilation and air conditioning. John Howatt, past president of the Society, presided and introduced the various guest speakers among whom were, Elmer Jensen, president of the *Illinois Society of Architects*, G. C. Gearon, Department of Boiler Inspection of Chicago, J. J. Aeberly, Chicago Board of Health, John Neale, Underwriters Laboratories, and J. T. W. Babcock, Western Factory Insurance. The meeting was attended by 124 members and their guests.

The Chapter also sponsored, with other groups, an Air Conditioning Symposium at which some of the best known authorities in the field of air conditioning gave a series of talks. Attendance at this meeting was nearly 700, a large number of whom were Illinois Chapter members.

All meetings were held at the Merchants' & Manufacturers' Club throughout the season. Thanks are due W. H. Howe for supplying and operating the projection machine and the fine newspaper publicity given to the Illinois Chapter activities this year has been due to the good work of V. L. Sherman.

The Chapter sent its Secretary to the Semi-Annual Meeting of the Society at Hot Springs and to the Annual Meeting in Pittsburgh. At both these meetings Illinois Chapter was well represented by members and their families. At the Annual Meeting, Tom Brown and M. W. Bishop served as member and alternate, respectively, from Illinois Chapter on the Society's Nominating Committee. J. R. Vernon, as President, was the Chapter's official delegate to this meeting.

The Board of Governors wish to express appreciation of the whole-hearted support accorded by the entire membership throughout the year.

May 15, 1939. The 33rd annual meeting of the Illinois Chapter was held at the Merchants' and Manufacturers' Club in the Merchandise Mart, with an attendance of 98 members, ladies and guests.

After dinner, the business session was called to order by President Vernon, who dispensed with the regular order of business, to devote his attention to the annual reports of officers and committee chairmen.

Following a brief introduction by President Vernon, J. F. McIntire, President of the Society, spoke to the Chapter, mentioning the past presidents who had been given to the Society by the Illinois Chapter, Messrs. Hale, Hart, Hill, Lewis and Howatt. He then sketched his nine weeks' tour through the west and gave some interesting sidelights on the Birth of Air Conditioning.

H. G. Chapin gave the report of the Tellers of the annual election, declaring that the following slate had been duly elected for 1939-40:

President—Tom Brown.
Vice-President—V. L. Sherman.
Secretary—M. W. Bishop.
Treasurer—E. M. Mittendorff.
Board of Governors—I. E. Brooke, W. A. Knechenberg, A. O. May and J. R. Vernon, past president.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 17 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by October 16, 1939, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BARTON, E. H., Owner & Mgr., Barton Agencies, Moosomin, Sask., Can.	William Worton J. B. Steele	P. L. Charles William Glass
CAMPBELL, R. P., Sales Engr., E. K. Campbell Heating Co., Kansas City, Mo.	E. K. Campbell J. L. Peltzman	K. M. Stevens Carl Clegg
DEMETER, Julius, Htg. & Air Cond. Engr., Julio Donoso D., Santiago, Chile.	F. G. Walther J. D. Constance	L. T. Bourgarde (*ASRE*) D. J. Murphy (*Non-Member*)
FIRESTONE, M. T., Mgr., Carrier Air Cond. Div., Washington, D. C.	Richard Bozeman E. S. de Witt	H. H. Downes T. H. Urdahl
GAIR, K. B., Sales Engr., B. F. Sturtevant Co., Grand Rapids, Mich.	C. A. Pavey W. H. Rose, Jr.	G. H. Cummins E. H. Clark
HALL, C. H., Chief Engr., Fairbanks, Morse & Co., Three Rivers, Mich. (*Reinstatement*)	A. E. Bastedo F. W. Hanburger	A. J. Johnson L. C. Bosler (*ASME*)
HARTSOOK, G. S., Jr., Mgr., Miller & Anderson, Front Royal, Va.	T. C. Cooke W. M. Wallace, II	H. L. McDowell C. Z. Adams
HIGH, J. M., Mgr., Insulation Div., The Ruberoid Co., New York, N. Y. (*Advancement*)	W. J. Osborn R. V. Sawhill	A. F. Hinrichsen W. E. Heibel
HILLEGER, M. L., Draftsman & Estimator, W. S. Kilpatrick & Co., Los Angeles, Calif.	W. S. Kilpatrick O. W. Ott	H. M. Hendrickson A. J. Hess
KARSUNKY, W. K., Consulting Engr., Washington, D. C.	P. H. Erisman, Jr. W. R. Lockhart	T. H. Urdahl F. A. Leser
NEAL, J. P., Mech. Engr., Fosdick & Hilmer, Cincinnati, Ohio.	I. B. Helburn E. B. Royer	H. E. Sproull R. E. Kramig, Jr.
NOLAN, J. J., JR., Air Cond. Engr., United Clay Products Co., Washington, D. C.	E. S. de Witt H. H. Downes	W. E. Kingswell T. H. Urdahl
ROGERS, T. L. C., Managing Dir. & Chief Engr., National Air Cond. Co., Ltd., London, Ont., Can.	J. G. O'Flaherty R. H. Everest	A. W. LeBel (*Non-Member*) H. Shaw (*Non-Member*)
ROSEN, E. J., Mgr., Temp. Control Dept., Parker-Carpenter, Inc., San Francisco, Calif.	R. P. Dewey G. M. Simonson	R. A. Parker C. E. Bentley
SCOTTI, F. D., Tech. Dir., G. De Micheli & C.S.A., Florence, Italy.	Luciano Dell 'Orto Giuseppe Marzorati	Aldo Gini A. V. Hutchinson
TUTSCH, R. J., Asst. Mgr., Iron Fireman of Milwaukee, Inc., Milwaukee, Wis.	F. G. Weimer H. W. Schreiber	J. H. Volk A. B. Meeg (*ASME*)
WATT, R. M., JR., Lt. Commander, Construction Corps., U. S. Navy, Washington, D. C.	A. E. Stacey, Jr. W. L. Fleisher	F. C. Houghten T. H. Urdahl

CANDIDATES ELECTED

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ANSPACHER, T. H., Dist. Mgr., Buffalo Forge Co., Dallas, Tex. (*Advancement*)

BISHOP, J. A., Dist. Mgr., American Blower Corp., Dallas, Tex.

MITCHELL, A. E., Mgr. Oil Burner & Air Cond. Dept., A. P. Woodson Co., Washington, D. C.

MURHARD, K. R., Sales Engr., U. S. Radiator Corp., Portland, Ore.

PELLEGRINI, L. C., Vice Pres. & Design Engr., Marlo Coil Co., St. Louis, Mo.

RIDLEY, W. H., Mgr. Dryer Div., Riggs & Lombard, Inc., Lowell, Mass.

ASSOCIATES

CLARK, R. L., Pres., Clark Asbestos Co., Cleveland Heights, Ohio (*Reinstatement*)

FLANAGAN, J. B., Sales Mgr., Warden-King, Ltd., Montreal, Que., Can.

HANSON, L. C., Htg. & Plbg. Contractor, Bjorkman Bros. Co., Minneapolis, Minn. (*Reinstatement*)

HAZLEHURST, H. D., Serviceman, Southern Calif. Gas Co., Los Angeles, Calif.

JONES, J. T., Power Sales Engr., Empire Dist. Elec. Co., Joplin, Mo.

ODUM, R. A., Supt. Htg. Dept., Grove C. Odum & Co., Daytona Beach, Fla.

SHARP, J. E., Htg. & Air Cond. Engr., Springfield Stoker & Htg. Co., Springfield, Ill.

STEVENS, W. H., Engr. & Service Mgr., Shellenberger, Gregg & Co., Milwaukee, Wis.

WILCOX, C. M., Design Engr., War Dept., Washington, D. C.

WILSON, W. E., Owner, Wilson Plumbing Co., Asheville, N. C.

JUNIORS

BARTELS, E. M., Supervisor Mech. Equip., Independent School Dist., Des Moines, Ia.

BURGES, J. H. M., Chief Draftsman, Lipscombe Air Cond. Co., Ltd., London, England.

BURNS, H. J., Asst. Htg. Engr., Washington Gas Light Co., Washington, D. C.

DADDARIO, F. T., Air Cond. Engr., The Lancaster Co., Johnson City, Tenn.

KAHN, C. R., Jr., Engr., Pabst Air Cond. Corp., New York, N. Y.

KOSTER, H. H., Asst. Prof., George Washington Univ., Washington, D. C.

NORFOLK, L. W., Consulting Engr., E. G. Phillips, Son & Norfolk, Nottingham, England.

STUDENTS

MATHEKA, C. R., Student, New York Technical Inst., New York, N. Y.